THE HIGH-SPEED COMPRESSION-IGNITION ENGINE

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PREFACE

The purpose of this book is not to describe the different makes of high-speed compression-ignition engine now being produced in such large numbers throughout the world, but to deal with the principles governing the operation of these engines and to discuss the problems associated therewith.

In using the term "high-speed" the author has in mind speeds such as are required for road transport engines. It is with engines falling into this class that the author's experience has been gained, and it is mainly from this point of view that the book has been written.

The high-speed engine is not the offspring of the heavy slow-speed engine, and in actual fact owes it little beyond the incentive to obtain for the small fast-running engine the improved economy obtained with the large engine when using a high ratio of expansion. In this country the imposition of the tax upon petrol was a severe blow to the heavy road transport business, and the possibility of using a tax-free fuel was an incentive added to the lure of improved fuel consumption. The monetary saving offered was considerable and, even with the relatively poor fuel consumption of the early engines, amounted to some £250 per annum in the case of a double-decked omnibus running in a city service. A saving of this magnitude was quite sufficient to make the large operator prepared to put up with the troubles inevitably associated with the development period of such an undertaking, and it was hoped that the period of freedom from fuel tax would be long enough to enable the worst of the troubles to be overcome. This hope was realized, and to-day, competing with the petrol engine on a basis of an equal basic cost of fuel, the high-speed compression-ignition engine has almost completely ousted the petrol engine from the heavier forms of road transport.

In adapting the principles of fuel injection and compression-ignition to engines for the road transport field a number of problems had to be tackled; the process of combustion had to be speeded up to a

rate commensurate with the rotational speeds required for automobile engines, while questions of smoke, smell and noise became problems of first-class importance on the public roads. To-day the process of combustion in the compression-ignition engine occupies less than half the time required by a petrol engine running at the same speed, and though the degree of success attained in the solution of the latter three problems may leave something still to be desired, it must be emphasized that development work is being carried on with a view to effecting further improvements.

In the course of its development the compression-ignition engine has had one tremendous advantage, one which is enjoyed by practically all developments in the automobile field, namely, the engineer's opportunity to accumulate mass experience at a very rapid rate. The individual machines being relatively small and used in large numbers, it is possible to place considerable numbers in service at a comparatively small cost, so that a large amount of practical experience is gained in a short space of time. Such a procedure is impossible with the large slow-speed engine, because not only is the demand strictly limited, but their high individual cost makes it imperative to proceed with caution, and the process of development is necessarily much slower. This does not mean that the development of the high-speed engine has been inexpensive; far from it. The cost has been tremendous, and it is due to the courageous attitude taken by the heads of the commercial-vehicle industry that the development of the high-speed engine has been carried to its present position within the space of ten years.

Although workers on the high-speed engine have not been able to draw upon the experience gained with the low-speed engine, the converse is by no means true. The knowledge gained from the intensive study of combustion problems and of the phenomena occurring in the injection system is of equal value to engines of all sizes and types, and all branches of the industry are taking advantage of the information now available.

The author's thanks are due to the Management of The Associated Equipment Co., Ltd., for the broad-minded view taken as to the publication of technical information. Without this attitude on their part the book could not have been written. He has also to express his thanks to Mr. H. R. Ricardo for his encouragement and also for certain information; to the Institution of Mechanical Engineers, the

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C. B. D.

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CHAPTER I

The Laws of Gases

The compression-ignition engine is governed by the same fundamental laws of gases as all other types of heat engines. We do not, however, propose to go exhaustively into these laws, but to deal with them only to the extent necessary to enable the student to obtain a clear understanding of the subject in so far as the operation of the compression-ignition engine is concerned. Those who desire further details are recommended to consult one of the numerous works on the subject.

1. Relationship between P, V, and T.

In a given weight of gas, the changes which take place in pressure, volume and temperature, as measured on the *absolute* scale, have a definite relationship to each other, and if the change in any two of these conditions is known, that of the third may be determined from the fundamental equation

$$PV = RT,$$

where P is the pressure in lb./sq. ft. (absolute),
V the volume in c. ft.,
T the temperature in degrees absolute,* and
R a constant $= 96$ for air.

It follows that

$$\frac{PV}{T} = R = \text{constant},$$

and consequently

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2},$$

where P_1, V_1 and T_1 are the pressure, volume and temperature of a given weight of gas under a given set of conditions, and P_2, V_2 and T_2

* Centigrade degrees are used throughout this book, except where otherwise stated.

are the corresponding values for the same weight of gas under a different set of conditions.

The value of R is the difference in value of the two specific heats of the gas, the specific heat measured at constant pressure and the specific heat measured at constant volume, and represents the work done by the gas in expanding at constant pressure when heated through one degree. It is not always clear to the student why a gas should have two values for its specific heat, but if the large amount of expansion which takes place when a gas is heated under constant pressure conditions is kept in mind, the reason for, and the significance of, the two values will readily be seen.

Suppose that the mass of gas which we are considering is at atmospheric temperature, and is contained within a cylinder closed at one end by a fixed cover and at the other by a frictionless gas-tight piston. As the piston offers no resistance to movement, the pressures within and without the cylinder will be identical and equal to that of the surrounding atmosphere. On being heated, the gas within the cylinder will expand, and as the piston is frictionless it will be pushed outwards by the gas and the pressures within and without the cylinder maintained equal as before. In moving outwards, however, the piston has done a certain amount of work in forcing back the atmosphere which presses against its outer surface with a pressure amounting, under standard conditions, to as much as 2116·8 lb./sq. ft.

Let us now suppose that the piston, instead of being free to move, is fixed so that the volume of the gas remains unchanged during the heating process, but, as in the first case, the pressure before heating is that of the atmosphere. On being heated, the gas now does no work against the outer atmosphere, although the pressure of the gas itself will rise, since it is being prevented from expanding.

In heating the gas under the first set of conditions, the amount of energy added to the gas consists not only of the heat required to raise the gas to the required temperature, but of the energy required to push the piston back against the surrounding atmosphere also. Under the second set of conditions, as no external work has been done, we have only to supply the energy necessary to raise the gas to the required temperature. The amount of energy required to raise a unit weight of a gas through a given temperature range is thus less under constant volume conditions than under constant pressure conditions, the difference being the amount of work done upon the surrounding atmosphere.

These two values of specific heat of a gas, usually denoted by K_p and K_v, are of the first importance in internal-combustion engine work. Their difference and their ratio are factors which enter into practically all theoretical calculations. In point of fact, their importance is such that, as will be seen hereafter, the ratio K_p/K_v, commonly denoted by

γ, governs the efficiency which may be attained by any given engine. The value of γ for pure dry air at ordinary temperatures is 1·408, for convenience usually taken as 1·4; the latter is the figure used when dealing with the theoretical side of internal-combustion engine work, the working medium used by all ideal engines being assumed to be pure dry air.

2. Expansion of Gases.

A gas may expand or, since the action is reversible, may be compressed under either of the following conditions: (1) under constant pressure; (2) in accordance with the equation $PV^n = \text{constant}$.

3. Expansion under Constant Pressure.

Under these conditions heat is added, or extracted, at such a rate that the pressure remains unchanged throughout the process of changing the volume. This form of expansion has comparatively little application to the practical heat engine, being confined to a short period during which the fuel is being introduced into the cylinder in the true Diesel cycle.

4. Expansion in accordance with the Equation $PV^n = C$.

Expansion of this type may be divided into two classes: (i) expansion at constant temperature, which is known as Isothermal Expansion, and (ii) expansion during which heat is neither received from, nor lost to, any external source during the process, which is known as Adiabatic Expansion. It is this latter, or rather an approximation thereto, which plays the chief part in practical internal-combustion engines of all types.

5. Isothermal Expansion.

Under isothermal conditions, heat is added, or extracted, at such a rate that the temperature remains unchanged during the change in volume. Since the temperature remains unchanged, the general equation $PV = RT$ reduces to $PV = \text{constant}$. This is the equation of a rectangular hyperbola, and the pressure varies inversely as the volume, so that $P_2 = \frac{P_1 V_1}{V_2}$.

6. Adiabatic Expansion.

As has already been mentioned, adiabatic expansion, or rather an approximation to it, plays the chief part in practical internal-combustion engines. In this process the change takes place in accordance with the equation

$$PV^n = \text{constant},$$

where n is the ratio of the two specific heats which, as has already been mentioned, is usually denoted by γ.

Thus $$PV^{\gamma} = \text{constant}.$$

But for any gas $$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2},$$

whence $$P_1 T_2 = P_2 T_1 \frac{V_2}{V_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Also, since $$PV^{\gamma} = \text{constant},$$
$$P_1 V_1^{\gamma} = P_2 V_2^{\gamma},$$

whence $$\frac{P_1}{P_2} = \frac{V_2^{\gamma}}{V_1^{\gamma}} \quad \text{and} \quad \frac{V_2}{V_1} = \frac{P_1^{1/\gamma}}{P_2^{1/\gamma}};$$

also, $$P_2 = \frac{P_1 V_1^{\gamma}}{V_2^{\gamma}}.$$

Substituting for $\frac{V_2}{V_1}$ in equation (1), we have

$$P_1 T_2 = P_2 T_1 \frac{P_1^{1/\gamma}}{P_2^{1/\gamma}},$$

whence $$\frac{T_2}{T_1} = \frac{P_2^{1-1/\gamma}}{P_1^{1-1/\gamma}} = \left(\frac{P_2}{P_1}\right)^{1-1/\gamma}.$$

Similarly it may be shown that

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad \text{and} \quad \left(\frac{P_2}{P_1}\right)^{1-1/\gamma} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}.$$

It will be observed that with isothermal expansion $P_2 = P_1 \frac{V_1}{V_2}$, whereas with adiabatic expansion $P_2 = P_1 \frac{V_1^{\gamma}}{V_2^{\gamma}}$. Since $\gamma = K_p/K_v$ and $K_p > K_v$, γ must be greater than unity. Hence the change in pressure resulting from a given change in volume is greater when the expansion is adiabatic than when it is isothermal. The same is true of compression; a curve connecting pressure and volume will always have a steeper slope when the change is adiabatic than when it is isothermal.

This may be explained in a less technical manner as follows. Since heat must be extracted during isothermal compression in order to maintain the temperature constant, and since during adiabatic compression no heat is either added or extracted by any external agency,

it follows that the temperature at the end of the adiabatic compression is greater than for the same change in volume made isothermally. If the temperature is greater, the pressure must be greater. Conversely, during an isothermal expansion heat is added to maintain a constant temperature, while with adiabatic expansion no heat is gained or lost from an external source. The temperature, and therefore the pressure, at the end of an isothermal expansion is therefore greater than for the same change in volume made adiabatically.

That is, for a given change in volume the change in pressure is always greater when the change is made adiabatically than for the same change made isothermally.

CHAPTER II

The Idealized Cycles of Operation

It is not proposed to discuss all the purely theoretical cycles of operation, but only the idealized forms of those which are actually found in practical engines. These are three in number and are known as (1) the Constant Volume Cycle, (2) the Constant Pressure Cycle, and (3) the Mixed Cycle, which is a combination of the other two.

1. The Constant Volume Cycle.

This cycle of operations, which takes its name from the fact that the volume of the working charge remains unchanged during both the reception and the rejection of heat, is shown diagrammatically in fig. 1.

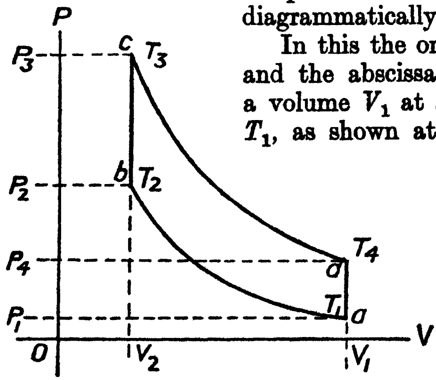


Fig 1

In this the ordinate OP represents pressures and the abscissa OV volumes. Starting with a volume V_1 at a pressure P_1 and temperature T_1, as shown at point *a* on the diagram, the gas is compressed adiabatically along the line *ab* to *b*, where the volume becomes V_2 and the pressure and temperature P_2 and T_2 respectively. While the volume remains constant at V_2 a quantity of heat is now added, thus increasing the temperature to T_3 and the pressure to P_3 at point *c*. Adiabatic expansion now occurs along the line *cd* to *d*, where the volume again reaches V_1, but at pressure P_4 and temperature T_4. While the volume remains constant at V_1, heat is now extracted until the pressure and temperature are restored to their original values of P_1 and T_1, and the cycle is ready to be repeated indefinitely.

If the heat which was added at point *b* is equivalent to H_2 heat units and that which was rejected at point *d* equivalent to H_3 heat

units, then the heat converted into useful work is $H_2 - H_3$ and the efficiency of the conversion is

$$E = \frac{H_2 - H_3}{H_2} = 1 - \frac{H_3}{H_2}.$$

If we consider unit weight of gas, the heat added, or lost, during any change in temperature which takes place at constant volume is given by the equation $H = K_v(T_x - T_y)$, where T_x and T_y are the temperatures, measured on the absolute scale, before and after the change in heat.

Hence
$$H_2 = K_v(T_3 - T_2)$$
$$H_3 = K_v(T_4 - T_1),$$

from which it follows that the efficiency E is given by

$$E = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}.$$

It has already been shown (p. 4) that with adiabatic expansion

$$\frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \frac{T_1}{T_2}.$$

Hence we also have

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{\gamma-1},$$

i.e.
$$E = 1 - \frac{T_4}{T_3} = 1 - \left(\frac{V_2}{V_1}\right)^{\gamma-1}.$$

V_1/V_2 is the ratio of the initial and final volumes of the gas. It is usually called the expansion ratio and is represented by the symbol r. Substituting this symbol in the foregoing equation, we have

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1},$$

or, as it is sometimes written,

$$E = 1 - \frac{1}{r^{\gamma-1}}.$$

Thus the efficiency of the ideal constant volume cycle depends upon the expansion ratio r and the ratio of the two specific heats γ. The efficiency is in no way influenced by the quantity of heat added or, what is really the same thing, the ratio of the pressures before and after the addition of the heat. The actual amount of work done will of course depend upon the quantity of heat added, but the efficiency

is entirely independent of it. This statement may be easily proved in the following manner.

If after compressing the gas adiabatically to V_2 at a pressure P_2 we add only a small quantity of heat sufficient to produce an increase in pressure equal to δp (fig. 2), the efficiency of the cycle by the above reasoning will be $E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$. If now instead of commencing our cycle at P_1 we begin at pressure P_x but still at volume V_1, and after compressing adiabatically to P_y at volume V_2 we again add a quantity of heat only sufficient to raise the pressure by an amount equal to δp, we can, by the same reasoning, prove that the efficiency of the cycle will be $E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$. The same holds good for any value of pressure P_x at which the cycle may be started, so that the efficiency of each little narrow strip corresponding to a pressure rise δp will be the same. The efficiency is thus independent of the quantity of heat which may be added * and, incidentally, of the pressure P_1 at which the cycle is started.

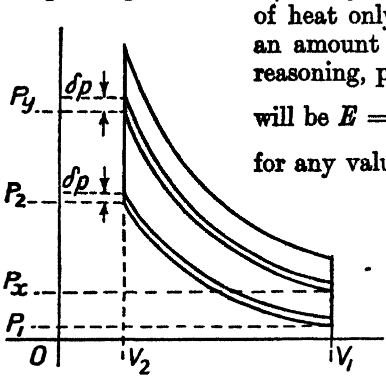


Fig. 2

When determining the efficiency of the constant volume cycle we thus have only two factors to consider, r and γ. For practical purposes this really reduces to one, the value of r, because, although the value of γ is a very influential factor, we are limited to the use of air as our working medium, and γ therefore assumes a known value, the ratio of the two specific heats of air. This for all practical purposes may be taken as 1·4. From the position of the factor $r^{\gamma-1}$ in the equation it will be observed that the greater the value of γ, the higher the value of E, but, unfortunately, it so happens that the method employed in practical engines for adding the heat (i.e. by the process of combustion) results in a reduction in the value of γ below the figure of 1·4 and a corresponding decrease in the maximum efficiency which it is theoretically possible to attain with a given value of r.

2. The Constant Pressure Cycle.

In this cycle the addition and extraction of heat are made while the pressure remains unchanged. The pressure volume diagram of the complete cycle is shown in fig. 3.

* As will be seen later, this statement has to be modified somewhat when considering practical working conditions.

Starting from point a, where conditions P_1, V_1 and T_1 prevail, the gas is compressed adiabatically to point b, where conditions P_2, V_2 and T_2 prevail. Here heat is added while expansion takes place, but without any change in pressure, till point c is reached, where the conditions are now P_2, V_3 and T_3. The supply of heat is then cut off and expansion takes place adiabatically to point d, at which the pressure has returned to P_1, but the volume is V_4 and the temperature T_4. Heat is now extracted without changing the pressure, which remains at P_1, until the volume returns to V_1 and the temperature to T_1.

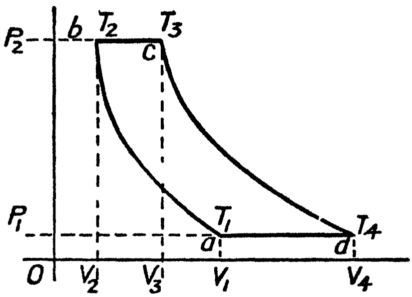


Fig. 3

As with the constant volume cycle, if the heat which was added between points b and c is equal to H_1 and that which was rejected between d and a to H_2, then the useful work done is represented by $H_1 - H_2$, and the efficiency is given by

$$E = \frac{H_1 - H_2}{H_1} = 1 - \frac{H_2}{H_1}.$$

Considering unit weight of gas, we have

$$H_1 = K_p(T_3 - T_2),$$
$$H_2 = K_p(T_4 - T_1),$$

and the efficiency is therefore

$$E = 1 - \frac{K_p(T_4 - T_1)}{K_p(T_3 - T_2)}.$$

Both expansion and compression being adiabatic, we have

$$\frac{T_4}{T_3} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_1}{T_2}$$

and

$$\frac{T_4}{T_3} = \frac{T_4 - T_1}{T_3 - T_2}.$$

Hence the efficiency becomes

$$E = 1 - \frac{T_4}{T_3} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}};$$

but with adiabatic expansion

$$\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{V_3}{V_4}\right)^{\gamma-1},$$

and therefore

$$E = 1 - \left(\frac{V_3}{V_4}\right)^{\gamma-1} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

This is the same expression as for the constant volume cycle, i.e. the constant pressure cycle and the constant volume cycle give the same efficiency for the same ratio of expansion.

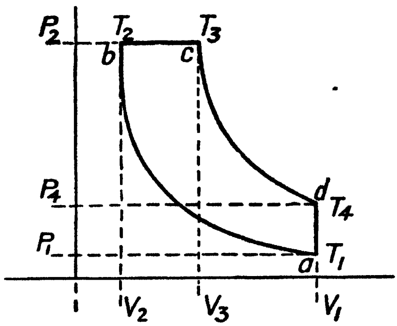


Fig. 4

With the constant pressure cycle as applied to actual engines, the expansion is not carried right down to the initial pressure P_1, but for practical reasons the expansion terminates when the volume has again reached V_1. Heat is then rejected at this point under conditions which, when idealized, may be considered as being under constant volume. The diagram for such a cycle, which is commonly called the Diesel Cycle, is shown in fig. 4.

Here again if H_1 represents the heat added between points b and c, and H_2 the heat rejected between d and a, the efficiency is given by

$$E = \frac{H_1 - H_2}{H_1} = 1 - \frac{H_2}{H_1}.$$

Since the heat H_1 is received at constant pressure,

$$H_1 = K_p(T_3 - T_2),$$

and since the heat H_2 is rejected at constant volume,

$$H_2 = K_v(T_4 - T_1).$$

From the general equation of gaseous relations,

$$\frac{P_2V_2}{T_2} = \frac{P_2V_3}{T_3},$$

and therefore $$T_3 = \frac{T_2V_3}{V_2},$$

so that $$H_1 = K_pT_2\left(\frac{V_3}{V_2} - 1\right).$$

Similarly $$T_4 = T_1 \times \frac{P_4}{P_1}.$$

From the equation for adiabatic expansion

$$P_1 = P_2\left(\frac{V_2}{V_1}\right)^\gamma \quad \text{and} \quad P_4 = P_2\left(\frac{V_3}{V_1}\right)^\gamma,$$

whence $$T_4 = T_1 \frac{P_2}{P_2} \frac{\left(\frac{V_3}{V_1}\right)^\gamma}{\left(\frac{V_2}{V_1}\right)^\gamma} = T_1\left(\frac{V_3}{V_2}\right)^\gamma.$$

Substituting this value for T_4 in the expression for H_2, we get

$$H_2 = K_v\left\{T_1\left(\frac{V_3}{V_2}\right)^\gamma - T_1\right\} = K_vT_1\left\{\left(\frac{V_3}{V_2}\right)^\gamma - 1\right\}.$$

But V_3/V_2 is the ratio of the volume at the end of heat reception and the volume at the end of compression, and may be designated by r_e. Substituting this symbol in the expressions for H_1 and H_2, we have

$$H_1 = K_pT_2(r_e - 1)$$
$$H_2 = K_vT_1(r_e^\gamma - 1)$$

and $$E = 1 - \frac{K_vT_1(r_e^\gamma - 1)}{K_pT_2(r_e - 1)} = 1 - \frac{T_1(r_e^\gamma - 1)}{T_2\gamma(r_e - 1)}.$$

The compression from V_1 to V_2 being adiabatic, we have

$$T_1/T_2 = 1/r^{\gamma-1},$$

as has been determined previously. Hence the equation for the efficiency of what is generally termed the Diesel cycle reduces to

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(r_e^\gamma - 1)}{\gamma(r_e - 1)}.$$

As will be seen, this equation contains the same minus factor as the equation for the constant volume cycle, but multiplied by a second factor $\frac{(r_e^\gamma - 1)}{\gamma(r_e - 1)}$. This second factor, which is governed by the ratio of expansion while the heat is being added, is obviously greater than unity. Further, the value of this second factor increases as the value of r_e increases, so that the *efficiency decreases as the period of heat reception is increased.* It follows, therefore, that for the same value of r the constant pressure cycle, as practically applied, is less efficient than either the true constant pressure cycle or the constant volume cycle, and that the decrease in efficiency will increase as the period of heat reception is increased.

The influence of r_e is clearly shown by fig. 5, which shows the value of E for values of r up to 25 and several values of r_e from 1 to 4. We see that for $r = 10$, the value of E is reduced from 60 per cent (to use round numbers) when $r_e = 1$ to 43 per cent when $r_e = 4$, a difference of some 28 per cent.

The difference between the two constant pressure cycles is well

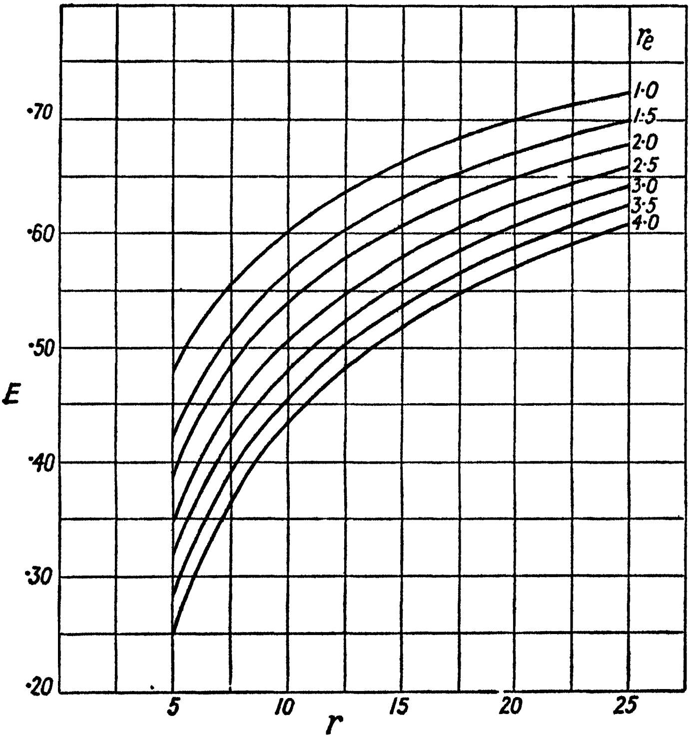


Fig. 5.—Efficiency of the constant pressure cycle for different ratios of expansion at constant pressure

illustrated by the combined diagram shown in fig. 6, which is lettered to correspond with figs. 3 and 4. The true constant pressure cycle expands right down till the pressure has fallen to its initial pressure P_1. To do this the volume has had to be increased to V_4 and the cycle has an efficiency $E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$. In its practical form, solely for mechanical reasons, the expansion is carried only as far as the initial volume V_1, the heat then being rejected at constant volume, and the work which would have been developed during the expansion from

V_1 to V_4 (represented by the shaded area in the diagram) is lost. The efficiency of the cycle is thereby reduced to

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(r_e^{\gamma} - 1)}{\gamma(r_e - 1)}.$$

This serves to illustrate very clearly the point, so frequently overlooked, that it is the expansion ratio which governs the efficiency of the engine, not the compression ratio. Also, it is the *expansion ratio measured from the point at which the heat is received* which is the governing factor. In other words—and this is very important in practical engines—the efficiency derived from each particle of heat depends upon the amount of expansion which takes place after that particle has been added. It follows, therefore, that if for any reason the expansion is curtailed, some loss in efficiency must result. At first sight it might appear that if the above statement is true, the true constant pressure cycle should be less efficient than the constant volume cycle, as heat is being added while the volume changes from V_2 to V_3 (fig. 3), so that the expansion ratio for heat which is added at, or just after, V_2 must be greater than that added just before V_3 is reached, and that the mean efficiency for each unit of heat must therefore be less than the maximum possible. This, however, is not the case, and it can be shown that for the true constant pressure cycle the expansion ratio is the same for each particle of heat, and, incidentally, that the expansion ratio is equal to the compression ratio.

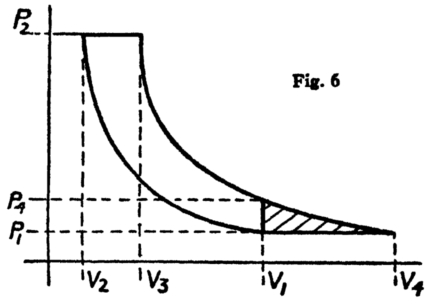


Fig. 6

On p. 4 we saw that

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{P_2}{P_1}\right)^{1-1/\gamma}.$$

We also have
$$\frac{P_1V_4}{T_4} = \frac{P_2V_3}{T_3};$$

hence
$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \left(\frac{P_2}{P_1}\right)^{1-1/\gamma};$$

but
$$\left(\frac{P_2}{P_1}\right)^{1-1/\gamma} = \left(\frac{V_1}{V_2}\right)^{\gamma-1},$$

therefore $$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} \text{ and } \frac{V_1}{V_2} = \frac{V_4}{V_3}.$$

That is, the expansion ratio from the termination of heat reception is the same as the compression ratio. The same line of reasoning may be applied to a small increment of heat during which the change of volume is so small as to be negligible, and it will be found that the whole diagram *a*, *b*, *c*, *d* may be considered as being composed of an infinite number of little strips, each of which is really a constant pressure diagram complete in itself, all having the same efficiency, so that the efficiency of the whole diagram is not altered by the fact that the heat is received while the volume is increasing from V_2 to V_3.

If now we apply the same line of reasoning to the cycle in its practical form with heat extraction at constant volume, we shall at once see why this cycle is inherently less efficient than the true constant pressure cycle.

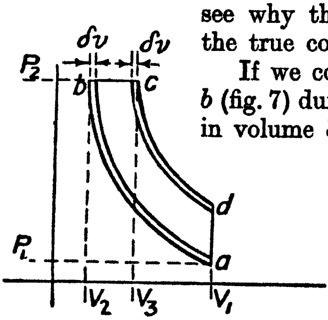


Fig. 7

If we consider a particle of heat added at point *b* (fig. 7) during the addition of which a small change in volume δv takes place, the expansion which subsequently occurs is given by V_1/V_2, whereas the expansion which takes place after a particle of heat has been added at *c* (fig. 7) can only amount to V_1/V_3, with a corresponding loss of work done and therefore of efficiency. Each succeeding particle added after the first particle at *b* will have a progressively smaller expansion following it, and a progressively smaller efficiency. Thus the longer the period of heat reception the lower will be the efficiency of the Diesel Cycle, as is clearly indicated by the expression for the efficiency of this cycle.

This point is of considerable importance in all practical engines, whether operating on the constant pressure cycle or the constant volume cycle, since the reception of heat is never exactly in accordance with the assumption made when analysing the theoretical cycle.

In a practical engine the heat is added by a chemical process called combustion and cannot be controlled with such exactitude that just the right amount of heat is liberated at every point in the cycle. The result is that the period of heat reception is always longer than theoretical conditions call for and the efficiency of the cycle suffers. Under practical operating conditions this drawn-out heat reception results in a material reduction in the effective expansion ratio and is one of the chief reasons why the efficiency falls short of what is theoretically possible.

3. The Mixed Cycle.

Under high-speed conditions practically all internal-combustion engines, compression-ignition, petrol or gas engines, operate on a cycle which is really a mixture of the constant volume cycle and the constant pressure cycle. In petrol and similar engines, this is due to the time which elapses before the combustion is anything like completed. The piston has been moving forward during this period so that an increase in volume has taken place, and while the pressure may not actually have remained constant, for a short period of expansion a condition approximating to constant pressure conditions has been produced.

For a compression-ignition engine to operate on the constant pressure cycle, combustion must commence at the inner dead centre of the piston and continue for a period at such a rate that the pressure does not rise above the compression pressure. If, however, attempts are made to operate a high-speed engine on this cycle, or anything even approximately resembling it, the time interval which elapses between the moments at which injection commences and the ignition of the fuel takes place results in so much delay in the burning of the fuel that a very serious loss of efficiency takes place. Not only this, but the combustion is so uncertain and incomplete that missfiring and a very dirty exhaust are produced.

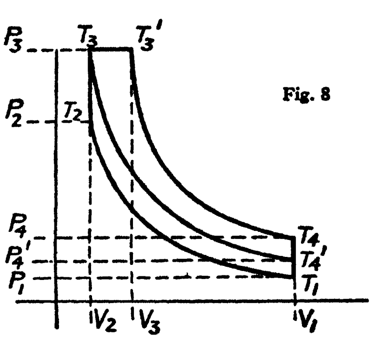


Fig. 8

This time interval, which is known as the "delay period", plays a very important part in the operation of high-speed compression-ignition engines, and is dealt with fully later on.

Advancing the injection timing so that ignition does take place at the inner dead centre, or more nearly so, does not eliminate this delay, and the presence of the fuel which has been delivered into the combustion chamber during the delay period results in the combustion of part, at least, of the fuel taking place at constant volume.

If ignition occurs at the inner dead centre, the combustion of any fuel which is admitted subsequently will take place under conditions of increasing volume, and may therefore be said to take place under conditions favourable to constant pressure operation. The idealized form of the true cycle of operation may therefore be said to consist of a period of constant volume combustion followed by a period of constant pressure combustion, as shown diagrammatically in fig. 8.

Starting at P_1, V_1 and T_1, the gas is compressed adiabatically to P_2, V_2 and T_2, at which point heat is added at constant volume V_2 to increase the pressure and temperature from P_2 and T_2 to P_3 and T_3. If no more heat were added, expansion would take place from P_3 until the original volume was again reached at V_1, expansion having taken place along the line to pressures P_4' and T_4' at V_1. The engine would then be working on the constant volume cycle and would have an efficiency given by the expression

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

If, however, further heat is added so as to maintain the pressure constant at P_3 while the volume increases from V_2 to V_3 and the temperature increases from T_3 at V_2 to T_3' at V_3, to be followed by adiabatic expansion from V_3 down to V_1 and temperature and pressure T_4 and P_4, we shall have a constant pressure cycle represented by the points T_4', T_3, T_3', T_4. The fact that the starting-point of this cycle, represented by V_1, P_4', T_4', is at elevated pressure and temperature as compared with the starting-point of the whole cycle, P_1, T_1, V_1, does not in any way modify the conditions governing the efficiency of the constant pressure cycle, so that the efficiency derived from the constant pressure part of the cycle is given by the expression

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(r_e^{\gamma} - 1)}{\gamma(r_e - 1)}.$$

The efficiency of the whole cycle thus depends upon the proportions of the heat supplied under the two conditions. Hence if H_v is the heat supplied under constant volume conditions, H_p the heat supplied under constant pressure conditions, and E_v, E_p are the corresponding efficiencies, the efficiency of the whole cycle will be given by the expression

$$E = \frac{H_vE_v + H_pE_p}{H_v + H_p} = \frac{H_v\left[1 - \left(\frac{1}{r}\right)^{\gamma-1}\right] + H_p\left[1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(r_e^{\gamma}-1)}{\gamma(r_e-1)}\right]}{H_v + H_p}$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{H_v + H_p\frac{(r_e^{\gamma}-1)}{\gamma(r_e-1)}}{H_v + H_p}.$$

H_v, the heat added at constant volume, is given by

$$H_v = K_v(T_3 - T_2),$$

and H_p, the heat added at constant pressure, by

$$H_p = K_p(T_3' - T_3) = \gamma K_v(T_3' - T_3).$$

Further,

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} = p, \text{ say,}$$

and

$$T_2 = T_1 r^{\gamma-1}.$$

$$\therefore\ T_3 = pT_2 = pT_1 r^{\gamma-1}.$$

Again,

$$T_3' = T_3 \times \frac{V_3'}{V_2} = T_3 r_e$$

$$= pT_1 r^{\gamma-1} r_e.$$

Hence

$$H_v = K_v(pT_1 r^{\gamma-1} - T_1 r^{\gamma-1})$$

$$= T_1 r^{\gamma-1} K_v(p-1).$$

$$H_p = \gamma K_v(pT_1 r^{\gamma-1} r_e - pT_1 r^{\gamma-1})$$

$$= (pT_1 r^{\gamma-1})\gamma K_v(r_e - 1).$$

$$\therefore\ E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{T_1 r^{\gamma-1} K_v(p-1) + (pT_1 r^{\gamma-1})\gamma K_v(r_e - 1)\dfrac{(r_e^{\gamma} - 1)}{\gamma(r_e - 1)}}{T_1 r^{\gamma-1} K_v(p-1) + (pT_1 r^{\gamma-1})\gamma K_v(r_e - 1)}$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(p-1) + p(r_e^{\gamma} - 1)}{(p-1) + p\gamma(r_e - 1)}$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1} \left[\frac{(pr_e^{\gamma} - 1)}{(p-1) + p\gamma(r_e - 1)}\right].$$

The expression for the efficiency of the mixed cycle has thus been reduced from one which contained an unknown factor, the distribution of the heat between the two types of combustion, to one depending upon the ratio of pressure rise during the constant volume part of the combustion and the amount of expansion which takes place during the constant pressure combustion.

This equation, which is due to D. R. Pye,* is really the most important of the several equations which give the efficiency of the different cycles, since not only does it give the efficiency of the idealized form of the cycle upon which all high-speed engines and also many other engines operate, but it is a universal expression giving the efficiency of the ideal form of all practical cycles. For an engine working entirely on the constant volume cycle, the value of r_e is 1, and if we substitute this value in the equation, that part of the

* The Limits of Compression Ratio in Diesel Engines, *Aeronautical Research Committee R. and M.*, No. 1365.

expression contained in square brackets becomes equal to unity and the whole expression reduces to

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1},$$

the expression for the constant volume cycle. Similarly, when an engine is working on the constant pressure cycle, we have $p = 1$, and the expression reduces to

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{(r_e{}^{\gamma} - 1)}{\gamma(r_e - 1)},$$

which is the expression previously derived for the Diesel cycle.

Apart from this, however, this equation enables us to determine the effect upon the efficiency of the engine when a limit has to be set to the maximum pressure which is developed in the engine. It is obvious that as applied in actual practice the maximum efficiency attainable for any given value of r is given by the constant volume cycle. For a given value of r the efficiency both of the mixed cycle and the practical constant pressure cycle is inherently less than that obtainable from the constant volume cycle, because the efficiency of both the other two cycles is obtained by first multiplying the minus factor of the equation by an additional factor, the value of which is always greater than unity. Any limitation imposed on the maximum pressure (other than in cases where it is produced by limiting the maximum quantity of heat which is added) must involve some portion of the heat being burned under conditions other than at constant volume, and therefore involves some reduction in efficiency. This loss of efficiency obviously results in some reduction in the quantity of work produced; but this loss of work will be very much less than that which would result from working wholly on the constant volume cycle and limiting the maximum pressure by a reduction in the quantity of heat added. Reference to fig. 8 (p. 15) will make this point clear.

A further point which is brought out by this expression is that with the mixed cycle the efficiency will tend to increase as the total quantity of heat added is decreased. A reduction in the quantity of heat added leads first of all to a reduction in the heat added at constant pressure, and therefore increases the proportion of the heat which is added under constant volume conditions and produces a corresponding improvement in efficiency. This improvement will increase as the quantity of heat added is reduced, until the point is reached where no heat is added under constant pressure conditions and all the heat is added under constant volume conditions. The efficiency attained at this point is that due to constant volume conditions alone, and therefore represents the maximum attainable.

Any further reduction in the quantity of heat added produces no further change in the conditions under which the heat is added and consequently produces no further change in efficiency. A reduction in the quantity of heat below the point at which constant pressure conditions disappear produces a steady decrease in the maximum pressure, until at zero heat the pressure does not rise above the compression pressure.

4. Conditions assumed for Idealized Cycles.

In arriving at an expression for the efficiency of the idealized forms of the cycles of operation certain conditions have been assumed. These conditions, not one of which is fully realized in the practical engine, are as follows:

1. The working medium has been assumed to consist of pure dry air, which in addition is assumed to be a perfect gas and to have a constant value for its specific heat.
2. No account has been taken as to how the heat is to be added or extracted, while the rates of extraction and addition are assumed to take place exactly as required, i.e. instantaneously for constant volume working and at such a rate that the pressure remains absolutely constant for constant pressure working.
3. Apart from intentional changes in heat, no heat is either gained or lost during the cycle.
4. The operation of the engine is frictionless.

The divergences of practical engines from the ideal engine are discussed at length hereafter, but it will not be amiss to summarize the chief ones here.

1. The working medium is not pure air and during the most important part of the cycle contains appreciable quantities of gases which differ widely from air. The specific heat is far from constant, and the properties of the working substance differ considerably from those of a perfect gas.
2. The heat is added by the process of combustion, which is never instantaneous, not always complete, and difficult to regulate so as to give true constant pressure conditions.
3. A loss or gain of heat at other than the designed points in the cycle is unavoidable.
4. Finally, the engine is by no means frictionless.

CHAPTER III

The Chemistry of Combustion

Combustion is the reaction between a combustible substance, termed the fuel, and the oxygen which is contained in the atmosphere. It is a definite chemical reaction and is accompanied by an evolution of heat, this heat being made use of to supply the energy necessary to produce the work required from the engine. Being a chemical reaction, combustion takes place in a well-defined manner and involves certain definite changes in the composition of the substances entering into the process, or rather, a rearrangement of their molecules.

1. The Fuel.

The fuels commonly available for internal-combustion engines of the liquid fuel variety are mineral hydrocarbon oils, which, as their name implies, consist of a combination of hydrogen and carbon molecules in proportions which vary somewhat according to the source from which the fuels are derived.

In addition to these two elements minute quantities of other elements also are found, notably oxygen and nitrogen, but in quantities which are small enough to be ignored in all but the most accurate physical investigation. Sulphur may be present also as an impurity in quantities as high as 2 per cent in some fuels; when present to this extent it should be taken into account when making any combustion calculations. In the fuels commonly used for high-speed engines in this country, however, the quantity of sulphur is sufficiently small to be ignored. For all practical purposes, therefore, we may assume that the fuel consists of hydrogen and carbon only.

2. The Atmosphere.

The atmosphere consists of a mixture of nitrogen and oxygen gases, together with minute quantities of certain other gases—quantities far too minute to need consideration. Water vapour is present also to an extent which varies considerably according to local climatic conditions, but in quantities such that it may as a rule be safely ignored.

For our purpose, therefore, we may assume that the atmosphere consists of a mixture of nitrogen and oxygen in the proportion of

76·85 parts per hundred by weight of nitrogen to 23·15 parts per hundred by weight of oxygen. By volume the proportions are 79·16 per cent nitrogen and 20·84 per cent oxygen.

Of the two gases, the oxygen alone enters into chemical reaction with the fuel; the nitrogen, being an inert gas, remains unchanged during the process of combustion. It therefore acts as a diluent to the oxygen and tends to slow down the rate at which the reaction takes place; by absorbing its share of the heat liberated, it reduces the maximum temperature, and therefore the maximum pressure, produced by the combustion.

3. The Chemical Reactions of Combustion.

The reaction between the hydrogen and the carbon of the fuel on the one hand, and the oxygen of the air on the other, takes place according to the following equations:

1. $$\underset{4}{2H_2} + \underset{32}{O_2} = \underset{36}{2H_2O} + \text{heat.}$$

2. $$\underset{12}{C} + \underset{32}{O_2} = \underset{44}{CO_2} + \text{heat.}$$

Expressed mathematically, these two equations mean that 4 parts, by weight, of hydrogen combine with 32 parts, by weight, of oxygen to form 36 parts, by weight, of water or water vapour, with an evolution of heat, and that 12 parts, by weight, of carbon, combine with 32 parts, by weight, of oxygen to produce 44 parts, by weight, of carbon dioxide, with an evolution of heat.

Carbon and oxygen can combine in yet another way:

3. $$\underset{24}{2C} + \underset{32}{O_2} = \underset{56}{2CO} + \text{heat,}$$

or 24 parts, by weight, of carbon combine with 32 parts, by weight, of oxygen to produce 56 parts, by weight, of carbon monoxide, with an evolution of heat. This reaction normally takes place when the quantity of oxygen present is insufficient to satisfy completely the requirements of the carbon. The carbon being incompletely consumed, the quantity of heat liberated is less than when the carbon is burned to carbon dioxide as in equation (2). This reaction is of comparatively small importance in the case of the compression-ignition engine, which normally has an appreciable excess of oxygen present. It is only under special circumstances, and then only in very small quantities, that carbon monoxide is found in the combustion products of the compression-ignition engine. In the case of carburettor engines which can conveniently use fuel/air mixtures which contain a con-

siderable excess of fuel, however, this reaction plays a very important part.

Carbon monoxide is itself a combustible gas, and the heat which has not been realized when the carbon burns to carbon monoxide may be released by adding further oxygen to the carbon monoxide in accordance with the following equation:

$$\underset{56}{2CO} + \underset{32}{O_2} = \underset{88}{2CO_2} + \text{heat.}$$

4. The Reactions between the Fuel and the Air.

In the previous section the reactions between the chemical substances hydrogen, carbon and oxygen were discussed, but while these are the reactions which enter into combustion problems in the engine, we do not under practical working conditions have to deal with the pure chemical elements, but with a fuel which is composed of hydrogen and carbon in chemical combination with one another, and atmospheric air, which is a mechanical mixture of the gases oxygen and nitrogen.

A fuel which consists of hydrogen and carbon, when completely burned to H_2O and CO_2, will produce the same quantity of each of these two substances as will the combustion separately of quantities of the two combustibles in an uncombined state equal to the amounts present in combination in the fuel.

The fuels normally available in this country for use in high-speed engines contain about 13 per cent of hydrogen and 87 per cent of carbon, and the complete combustion of 1 lb. of such a fuel will produce the same quantity of water vapour and carbon dioxide as will the combustion separately of 0·13 lb. hydrogen and 0·87 lb. carbon. As we have to resort to atmospheric air for our supply of oxygen and the atmosphere contains only 23·15 per cent by weight of this gas, we are compelled to take into consideration a much greater weight of atmospheric air than the actual weight of oxygen required. From equation (1) we find that 1 lb. of hydrogen requires $32/4 = 8$ lb. of oxygen for its complete combustion and from equation (2) that 1 lb. of carbon requires $32/12 = 2{\cdot}666$ lb. of oxygen for its complete combustion. But 1 lb. air contains only ·2315 lb. oxygen, so that 1 lb. hydrogen will require

$$\frac{32}{4} \times \frac{1}{\cdot 2315} = 34{\cdot}5572 \text{ lb. air,}$$

and 1 lb. carbon will require

$$\frac{32}{12} \times \frac{1}{\cdot 2315} = 11{\cdot}5191 \text{ lb. air}$$

for complete combustion.

A fuel which contains 13 per cent hydrogen and 87 per cent carbon will therefore require for complete combustion

$$\cdot13 \times 34{\cdot}5572 + \cdot87 \times 11{\cdot}519 = 14{\cdot}513966, \text{ say, } 14{\cdot}514 \text{ lb. air.}$$

The quantities of combustion products which will result from the complete combustion of this fuel will be as follows:

$$\cdot13 \times \frac{36}{4} = 1{\cdot}17 \text{ lb. of water vapour } (H_2O).$$

$$\cdot87 \times \frac{44}{12} = 3{\cdot}19 \text{ lb. of carbon dioxide } (CO_2).$$

In addition, there is the nitrogen contained in the original quantity of air supplied, which will have passed through the combustion process unaltered. This will amount to

$$14{\cdot}514 \times \frac{76{\cdot}85}{100} = 11{\cdot}154 \text{ lb. nitrogen } (N_2).$$

The total weight of products will, of course, be equal to the weight of air plus the weight of the fuel, i.e. there will be

$$14{\cdot}514 + 1 = 15{\cdot}514 \text{ lb. products.}$$

5. The Effect of an Excess of Air.

The foregoing represents conditions which result from the combustion of a mixture of fuel and air of such proportions that the whole of both the available oxygen and the combustibles of the fuel are completely used up, and the products therefore consist of water vapour, carbon dioxide and unchanged nitrogen. If an excess of air is present, the composition of the gases after combustion will be modified by the presence of the excess air which will have passed through the process unaltered. This means that for every pound of excess air there will be a pound of unused air mixed with the products of combustion, i.e. there will be 0·2315 lb. of unused oxygen and an additional 0·7685 lb. of nitrogen. It is thus easy to determine the composition of the mixture of gases which results from the combustion of any fuel/air ratio between that containing the theoretical mixture for making complete use of both fuel and air and one containing an infinitesimal quantity of fuel.

In compression-ignition engine work, the usual practice is to speak of the fuel/air ratio in terms of the quantity of excess air which is present, the excess being expressed as a percentage of the quantity of air required for the complete combustion of the fuel. This method of expression, and also the alternative form of pounds of air per pound of fuel, gives a very inconvenient form of relationship, which tends

towards infinity when the fuel charge is small. In point of fact, there is very little justification for expressing the relationship between the fuel and the air in this manner. The engine deals primarily with the air, and as much fuel as the air can consume is then added to it. The power obtainable from the engine depends not upon the fuel which can be got into it—there is no limit to the fuel which can be fed into the engine—but, firstly, upon the quantity of air which is taken in by the engine, and secondly, upon the use that can be made of the air which the engine has obtained. For these reasons the author prefers to express all matters relative to combustion in terms of the percentage of the available air which has been used. Expressed in these terms, all readings have very nearly a straight-line relationship, and their meaning is thus much more readily appreciated.

The convenience of this method will be made clear by the readings in Table I, which shows, both by weight and also as a percentage, the composition of the products of combustion formed when the fuel just discussed is burned with varying proportions of air.

TABLE I

Combustion products produced by burning various quantities of fuel in 14·514 lb. of air

Air used %	Fuel burned lb.	H_2O lb.	H_2O %	CO_2 lb.	CO_2 %	O_2 lb.	O_2 %	N_2 lb.	N_2 %	Total Wt. lb.	Excess Air %
100	1·0	1·17	7·53	3·19	20·56	—	—	11·154	Balance	15·514	0
90	·9	1·053	6·83	2·871	18·61	·336	2·18	11·154		15·414	11
80	·8	0·936	6·11	2·552	16 66	·672	4·39	11·154		15·314	25
70	·7	·819	5·38	2·233	14·68	1·008	6·62	11·154		15·214	42·85
60	·6	·702	4·64	1·914	12·66	1·344	8·89	11·154		15·114	66·67
50	·5	·585	3·89	1·595	10·62	1·680	11·18	11·154		15·014	100
40	·4	·468	3·14	1·276	8·56	2·016	13·52	11·154		14·914	150
30	·3	·351	2·37	0·957	6·46	2·352	15·88	11·154		14·814	233·3
20	·2	·234	1·59	0·638	4·33	2·688	18·27	11·154		14·714	400
10	·1	·117	0·801	0·319	2·18	3·024	20·69	11·154		14·614	900
0	0	—	—	—	—	3·360	23·15	11·154		14·514	∞

For the purpose of this table the quantity of air necessary for the complete combustion of 1 lb. of fuel is taken as a unit, and for the lower quantities of heat supplied the quantity of fuel has been taken as being reduced, as is the case in the actual engine, instead, as is usually the case, of assuming that the weight of the fuel remains constant and that of the air is increased. To show the inconvenience which follows upon the latter procedure the corresponding values of excess air are given. Figs. 9 and 10 show these figures plotted against the quantity of air used and also the excess air, and bring out very clearly the convenience of using the proportion of air used as a working basis. In fig. 9 the lines have a very slight curvature owing to the

fact that the total weight of the charge varies with the quantity of air used from a maximum of 15·514 lb. when 100 per cent of the

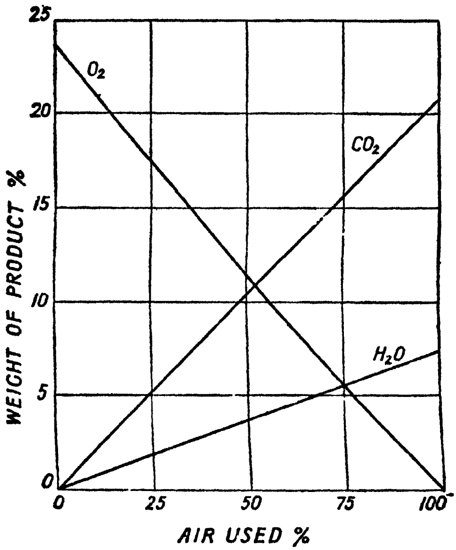


Fig. 9.—Composition of products of combustion plotted on a basis of air used

air is used, down to 14·514 lb. when zero air is used. The actual weights of the individual products when plotted on this basis give a perfectly straight line.

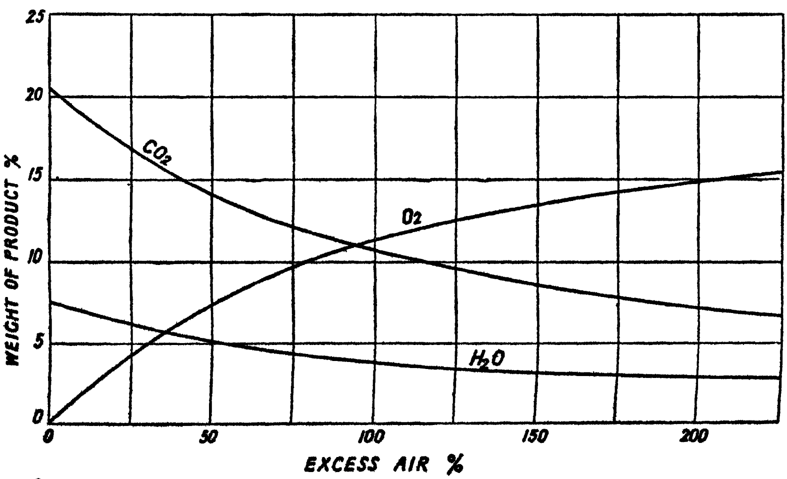


Fig. 10.—Composition of products of combustion plotted on a basis of excess air

If the relationship between CO_2 and O_2 in the combustion gases is plotted as shown in fig. 11, it is found that a true straight-line relationship exists, and from this diagram the proportion of air used may be determined by marking a suitable scale along the diagram as shown.

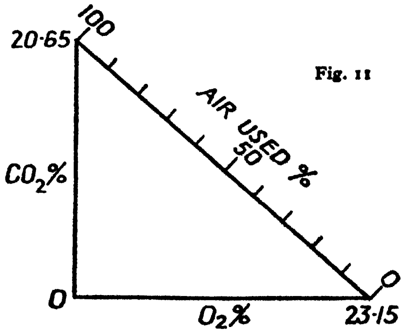


Fig. 11

6. Carbon Monoxide.

The presence of carbon monoxide in the exhaust of any engine represents a material loss of heat, so that, although this gas is not present in any appreciable quantities in the exhaust from a compression-ignition engine, such small quantities as may be produced should be taken into account when investigating the combustion efficiency of an engine. Under the conditions of combustion which exist in petrol and gas engines carbon monoxide and free oxygen do not appear together in the exhaust gases, but carbon monoxide is produced in considerable quantities when, as may easily happen in such engines, mixtures deficient in air are used. Actually, it takes a slight excess of air to eliminate all traces of carbon monoxide, this gas being present to the extent of about 0·4 per cent in the exhaust when the theoretical fuel/air mixture is used.

In compression-ignition engines, for reasons which are dealt with later, fuel/air mixtures which are deficient in air cannot be used, and in practice for satisfactory operation it is always found necessary to have a considerable quantity of excess air present. Despite the presence of free oxygen, small quantities of carbon monoxide are sometimes found; these are probably due to a local deficiency of oxygen which has not been corrected by the air movement until the temperature has fallen too low for combustion to take place.

It is a matter of considerable importance to be able to discover and eliminate the presence of even a small amount of carbon monoxide, but the accurate determination of this gas is a much more tedious and difficult matter than that of oxygen and carbon dioxide. The method of determination is the same in principle for all three gases, that is, by absorption, but the rate of absorption of carbon monoxide in its particular reagent is slow, and this, coupled with the very small quantities normally present in the exhaust of a compression-ignition engine, materially reduces the accuracy of the determination.

Provided that the correct relationship between O_2 and CO_2 is known for the fuel being used, the presence of carbon monoxide will

be indicated if the results from an analysis for these two gases are plotted on a diagram similar to fig. 11. The presence of carbon monoxide means that with a given fuel/air ratio the quantity of CO_2 will be *less* than would be the case had combustion been complete. It means also that the quantity of oxygen will be *greater* than that which would be found with complete combustion. This is due to the presence of an atom of free oxygen for every molecule of carbon monoxide. In chemical terms, 12 parts by weight of carbon have been used to produce 28 parts by weight of carbon monoxide instead of 44 parts by weight of carbon dioxide, i.e. for each 28 parts of monoxide 44 parts less of dioxide will be produced. Again, for every 28 parts by weight of carbon monoxide which have been produced there are 16 parts by weight more free oxygen than would have been the case

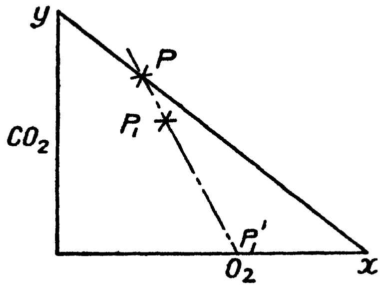


Fig. 12

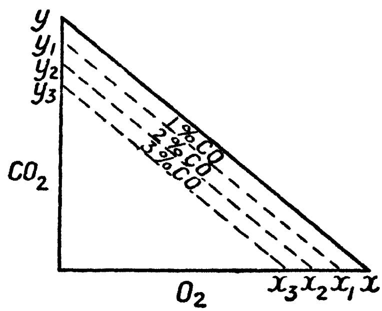


Fig. 13

had the combustion been completed. Reduced to more specific terms, for every 1 per cent by weight of carbon monoxide found there will be 1·5714 per cent less carbon dioxide and ·5714 per cent more oxygen than are produced by complete combustion.

If, therefore, any carbon monoxide is present, readings of O_2 and CO_2, when referred to a diagram like fig. 11, will fall diagonally below their correct position on the line xy as indicated at P_1 in fig. 12, P being the correct position for complete combustion.

From the reasoning just given it will be seen that the effect upon the quantity of O_2 and CO_2 of any given quantity of CO is independent of the quantity of O_2 and CO_2 and depends solely upon the quantity of CO which is formed. In other words, a given quantity of CO reduces the CO_2 content and increases the O_2 content of the combustion products to the same extent, no matter what the fuel/air ratio may be. We can, therefore, as shown in fig. 13, draw a series of lines x_1y_1, x_2y_2, . . . parallel to the original line xy of the diagram, fig. 12, to represent the position of the O_2–CO_2 line when, say, 1 per cent, 2 per cent, . . . of carbon monoxide is present, and from the resulting

diagram we shall be able to ascertain at once from an analysis giving only O_2 and CO_2, not only the presence of carbon monoxide, but also the actual quantity.

In fig. 11 a scale representing air used was marked out along the line *xy*, but this cannot be used directly if, owing to the presence of carbon monoxide, the results from an analysis do not actually fall on the line *xy*, but below it as at P_1 (fig. 12). The displacement of P_1 from its proper point P is, however, a linear function depending solely and directly upon the quantity of CO formed. If, therefore, a line is drawn from P through P_1 to meet the O_2 axis at P_1', all analyses of the exhaust products resulting from the combustion of the fuel/air ratio which with perfect combustion would give readings falling at the point P will, under conditions giving incomplete combustion, fall somewhere along the line PP_1'. The position along the line PP_1' at which the readings from any particular analysis falls will depend solely upon how much carbon monoxide is formed. For a reading to fall at P_1' it will be necessary for the whole of the available carbon to have been burned to carbon monoxide, since if it is to fall on the line *Ox* the amount of CO_2 must be zero.

Hence, in order to complete our diagram and be able not only to determine the completeness of combustion by ascertaining how much carbon monoxide is present, but also to determine what the exhaust analysis should be if combustion had been perfect, and at the same time arrive at the quantity of air which would have been used under conditions of perfect combustion, we need a series of parallel diagonal lines drawn from points along *xy* down to the *Ox* axis, such that each line will represent a constant fuel/air ratio. These lines may be drawn either by the use of the figures already determined, namely, that every 1 per cent of carbon monoxide results in 1·5714 per cent less CO_2 and 0·5714 per cent more O_2 than is produced by perfect combustion, or by assuming that for any given fuel/air ratio the whole of the carbon is burned to carbon monoxide and calculating the resulting products.

This method gives the maximum possible CO for the mixture in question, and a point representing zero CO_2 with a certain quantity of O_2. The most convenient fuel/air ratio to take is that which results in 100 per cent use of the fuel and air. By joining the point which corresponds to 100 per cent carbon burned to CO_2 to that which represents 100 per cent carbon burned to CO, we have a line on the diagram which gives the required slope of the diagonals. Lines drawn parallel to this line through suitable points of the air-used scale on the line *xy* will then complete the diagram and, in conjunction with the CO lines already obtained, enable us to obtain, from an analysis giving oxygen and carbon dioxide only, full information as to the degree of perfection of the combustion and the extent to which the available air has been used.

This form of combustion diagram, which is known as the Ostwald diagram, is very generally used on the Continent, but does not seem to be as well known in this country as its usefulness warrants. A diagram of this kind constructed on a volumetric basis, used along with the ordinary Orsat flue gas analysis apparatus, enables the experimental engineer or the operator of a power plant to detect any imperfection in the combustion of his engine in a way that no other apparatus can do.

7. The Composition of Products of Combustion on a Volumetric Basis.

Up to this point we have been dealing with the composition of the combustion products and their relative proportions based upon weight. In dealing with gases weight is a difficult unit to handle, and the simpler and much more usual method is to deal in volumes. In analysing a sample of exhaust products the procedure normally adopted is to measure out a definite volume of the gases and then to determine the quantity of each constituent present by removing each in turn by absorbing it in a suitable reagent. The volume of any desired constituent is given by the difference in volume of the sample before and after treatment by the proper reagent.

On account of the differing molecular weights of the several gases the proportions as measured on a volumetric basis differ materially from those determined from a weight basis. Also, owing to the fact that measurements on the volumetric basis are made at atmospheric temperature, one important constituent, H_2O, will be missing, having been condensed and absorbed in the water employed in the apparatus for drawing off and manipulating the sample. The disappearance of the water very naturally alters materially the proportions of CO_2, O_2 and N_2 which will be found in any given sample.

8. The Universal Gas Constant.

In determining the volumetric proportions of the combustion products from the proportions based upon weights, the law of Avogadro is used. Avogadro's law states that when measured under the same conditions of temperature and pressure equal volumes of gases contain equal numbers of molecules. In other words, the weights of such volumes are proportional to their molecular weights. This property of gases provides us with a constant known as the mol. A mol of gas is a weight such that if m denotes the molecular weight of the gas, one mol of the gas weighs m pounds (or kilograms, according to the units used). One mol of all gases contains the same number of molecules and consequently occupies the same volume. One mol of nitrogen is 28 lb. nitrogen; one mol of hydrogen is 2 lb. hydrogen, and so on; and one mol of nitrogen occupies the same

volume as one mol of hydrogen or of any other gas. The volume of one pound-mol of all gases when measured at 0° C. and 30 in. mercury is 358·7 c. ft. The following table gives the molecular weights, specific volumes, and specific weights of the various substances normally involved in the combustion reactions of a compression-ignition engine.

Gas	Chemical Symbol	Molecular Weight	Specific Volume c. ft./lb.	Specific Weight lb./c. ft.
Air	—	28·92 *	12·403	·080625
Oxygen	O_2	32	11·209	·08921
Nitrogen	N_2	28	12·81	·078064
Hydrogen	H_2	2	179·35	·005575
Carbon dioxide ..	CO_2	44	8·152	·12266
Carbon monoxide ..	CO	28	12·810	·07806
Steam †	H_2O	18	19·93	·050175
Carbon †	C	12	29·89	·033455

As a unit the mol has other uses, and for many gaseous calculations it is convenient to work in mols rather than in either weights or the more usual volumetric units.

In order to convert into volume a given weight of a gas, the weight of gas must be multiplied by its specific volume. In many instances, however, proportions are required rather than the actual volumes, and in this event it is only necessary to divide the weight of each gas present by the appropriate molecular weight. The answer obtained is the number of mols, or fraction of a mol, which the original weight represents. In dealing with exhaust gases, proportions are usually all that are required, so that this latter method is to be preferred, especially as molecular weights are much more readily committed to memory than specific volumes. If the figure 358·7 can be remembered as well, we are in possession of all we shall ever need to know about gases for the purpose of combustion calculations.

It must be remembered that volumetric proportions of exhaust gases can be expressed in two ways, one including the water vapour and the other on the assumption that the water vapour has been condensed and therefore occupies a negligible volume. The results obtained by the two methods will, of course, differ widely, and it is on the latter basis that exhaust gas analyses are usually made.

Using the method just outlined and taking the weights of gases

* This figure has no chemical significance, but is the equivalent figure for the mixture of gases forming the atmosphere.

† These two substances do not, of course, appear in the gaseous form under the conditions of temperature and pressure named, but the figures represent the values given by the universal gas constant and may be used in calculations where these substances appear.

which result from the combustion of the theoretical mixture for complete combustion, we arrive at the following:

Gas	Weight lb.	Mols of Gas	Volumes c. ft.	Composition %	
				with H_2O	with H_2O condensed
H_2O	1·17	·0650	23·32	12·14	—
CO_2	3·19	·0725	26·00	13·53	15·4
O_2	0·00	—	—	—	—
N_2	11·154	·3984	142·88	74·33	84·6

The series of fuel/air mixtures shown in Table I, but converted to a volumetric basis, is given in Table II, and is shown plotted upon a base of "air used" in figs. 14 and 15; fig. 14 includes the water vapour, while fig. 15 shows the proportion after the water vapour has condensed. Fig. 16 shows the corresponding Ostwald diagram based upon volumes.

TABLE II

Composition of Combustion Products by Volume

Air used %	Including H_2O			H_2O Condensed	
	H_2O %	CO_2 %	O_2 %	CO_2 %	O_2 %
100	12·13	13·70	0·0	15·40	0·00
90	10·99	12·48	1·97	13·76	2·22
80	9·84	10·78	3·98	12·15	4·40
70	8·65	9·65	5·99	10·55	6·55
60	7·46	8·32	8·02	9·00	8·68
50	6·26	6·98	10·11	7·44	10·77
40	5·03	5·61	12·20	5·92	12·82
30	3·80	4·24	14·32	4·41	14·90
20	2·55	2·84	16·48	2·86	16·57
10	1·28	1·43	18·65	1·45	18·89
0	0·00	0·00	20·84	0·00	20·84

N_2 Balance N_2 Balance

9. The Change in Volume after Combustion.

Any change in volume caused by the rearrangement of the molecules after combustion is a matter of considerable importance because it will directly influence the efficiency of the engine. By the phrase "change in volume" is meant a change in volume as recorded under the same conditions of temperature and pressure before and after combustion has occurred. With the gases confined within a container

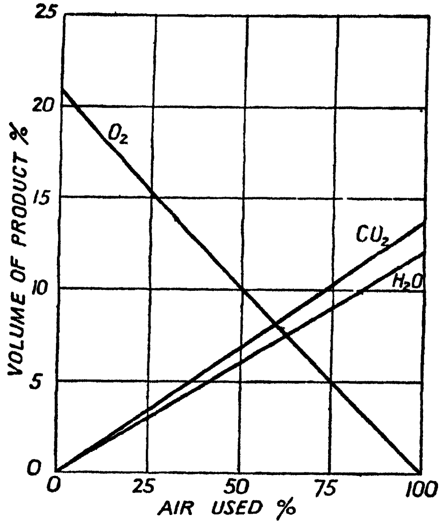


Fig. 14.—Composition by volume of products of combustion with H_2O as vapour

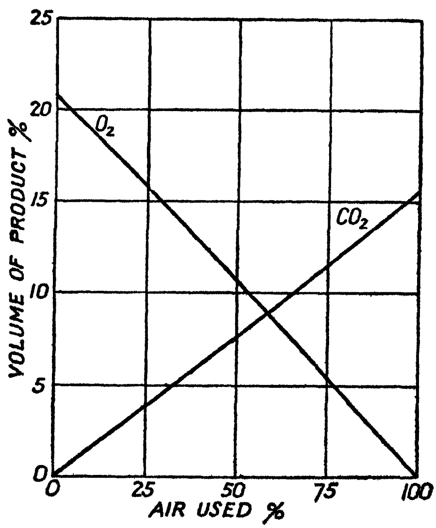


Fig. 15.—Composition by volume of products of combustion with H_2O condensed

of fixed volume, the absolute pressure of the gas will be influenced in direct proportion to any change in volume, measured as specified above, which may occur. Any such change in pressure will be recorded

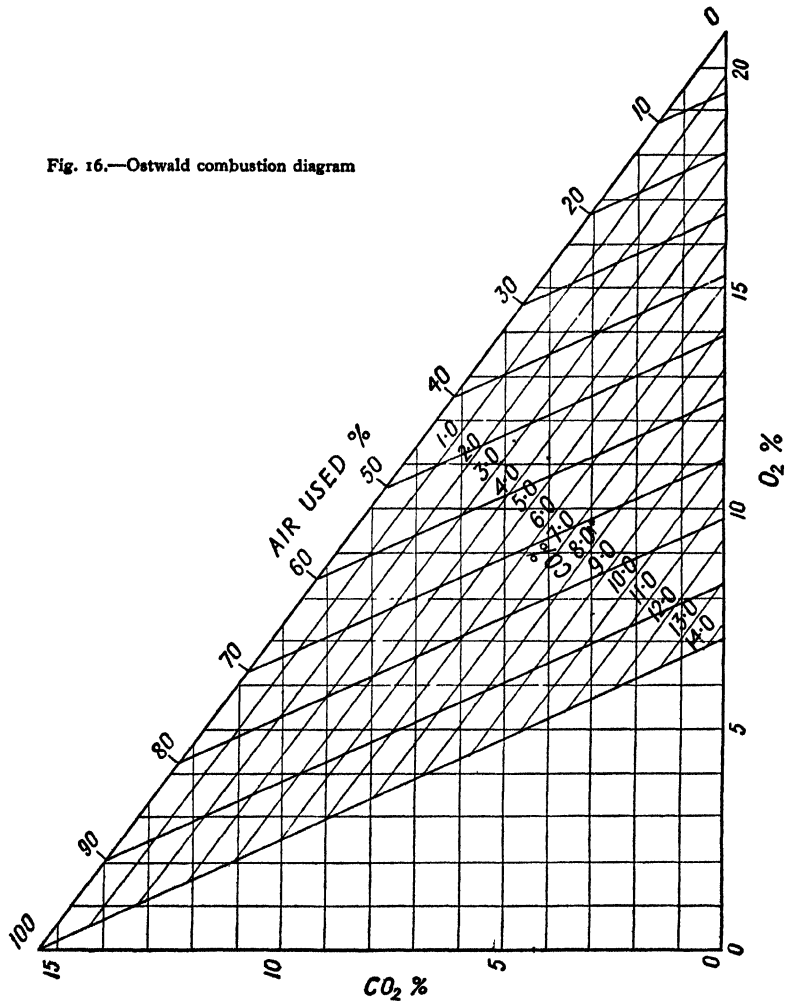


Fig. 16.—Ostwald combustion diagram

at all points of the expansion stroke and will have a direct effect upon the work done and therefore upon the efficiency of the engine.

It is important to ascertain what change, if any, takes place as the result of the combustion of 1 lb. of fuel with the theoretical weight of air. This, as already determined, involves the use of 14·514 lb. of air,

which under conditions of N.T.P. will occupy $14{\cdot}514 \times 12{\cdot}4 = 179{\cdot}97$ c. ft. The volume of the fuel which is added in liquid form may be neglected, since 1 lb. of a fuel such as is used for high-speed engines (S.G. ·85) will occupy a trifle less than ·02 c. ft. Before combustion, therefore, we have 14·514 lb. of gaseous matter occupying 179·97 c. ft. and 1 lb. of liquid occupying ·02 c. ft. (If vaporized, the fuel would occupy something of the order of 3·5 c. ft. measured at N.T.P.) After combustion, however, we have 15·514 lb. of gaseous matter—the water existing as vapour under all material circumstances—and the problem now is to determine the volume this mass of gas will occupy compared with the volume it occupied before combustion took place.

In determining the volumetric proportions of the products from the combustion of the theoretical mixture of fuel and air it was shown that we had the following:

1·17 lb. H_2O	which occupy	23·32	c. ft. at N.T.P.
3·19 lb. CO_2	,, ,,	26·00	,, ,,
11·15 lb. N_2	,, ,,	142·88	,, ,,
	Total	192·20	c. ft.

This is compared with an original volume of 179·97, giving an increase in volume of very nearly 7 per cent. Even allowing for the fuel in a vaporized form, the increase would amount to 5 per cent, but as the fuel is not admitted in this form the greater figure gives the correct increase.

With larger quantities of air, the excess air remains unchanged and the change in volume will vary directly with the quantity of air actually used, i.e. it will vary very nearly as a straight line from zero at zero air used, up to 7 per cent with 100 per cent air used, while the efficiency obtainable from the engine will benefit in the same proportion.

10. The Heat of Combustion.

The chemistry of combustion having been dealt with, we may now turn to the effects of the combustion in so far as it governs the supply of energy to the engine. From this point of view we are concerned with the following three reactions: (1) hydrogen burning to water; (2) carbon burning to carbon dioxide; (3) carbon burning to carbon monoxide.

Under conditions favouring complete combustion, the last of the three does not occur to any great extent, and, because of the excess of oxygen which is always present in compression-ignition engine work, carbon monoxide is not normally produced in anything but exceedingly small quantities. The reaction must, however, be considered in

order that the loss occasioned by the presence of even a small quantity of carbon monoxide may be evaluated.

Hydrogen burning to water liberates 34,142 lb. calories (Centigrade heat units, or C.H.U.) as measured at ordinary atmospheric temperature (15° C.). In cooling down to this temperature the water has been condensed, and has therefore liberated both the latent heat of evaporation of the steam which has been formed and also the sensible heat of the water between 100° C. and 15° C. These two quantities together amount to as much as 621 lb. calories per pound of water vapour, and as the combustion of 1 lb. of hydrogen produces 9 lb. water vapour, this represents the not inconsiderable quantity of 5589 lb. calories per lb. of hydrogen burned. What may be termed the available or useful heat from the combustion of hydrogen, therefore, amounts to 28,553 lb. cal./lb. This figure of 28,553 lb. cal./lb. represents the amount of heat per pound of hydrogen which is available after combustion to increase the energy contained by the gases in the engine cylinder, and is the figure used when investigating the performance of an engine from the theoretical standpoint. It will be obvious that for the purpose of calculating temperatures and pressures it is this lower calorific value which must be used, because the latent and sensible heats of the water vapour are not available until after a change of state of the water vapour has taken place. In assessing the performance of an engine from a commercial standpoint the gross or higher calorific value is now generally employed.

Carbon burning to carbon dioxide liberates the whole of its heat of 8067 lb. calories per pound, but in burning to carbon monoxide it liberates only 2406 lb. calories per pound of carbon; this latter reaction, therefore, results in only about 30 per cent of the total quantity of heat becoming available. The remaining heat is contained in the carbon monoxide gas, which is itself a combustible gas. The loss which accompanies the production of any carbon monoxide is thus apparent, and the necessity for avoiding the formation of this gas obvious.

The calorific value of carbon monoxide itself is 2436 lb. calories per pound, so that for every pound of this gas carried away in the exhaust gas there is a loss of heat to the extent of 2436 lb. calories.

As has already been said, the fuels available in this country may be considered as consisting of carbon and hydrogen only in the proportions of 87 per cent carbon and 13 per cent hydrogen. Small percentages of other substances are found in many fuels, sulphur being the commonest and also the one found in the greatest proportion. The sulphur does not appear in chemical combination, and is largely removed by the refining process undergone by the fuel. The maximum quantity of sulphur is limited by specification, not more than 2 per cent being allowed in the heavier fuels and 1·5 per cent in fuels

which are intended for high-speed engines by the B.E.S.A. Specification No. 209–1937, although about half this latter figure is the more usual quantity. Certain other elements are found also, notably oxygen and nitrogen, but in such small quantities that they may be ignored. For all practical purposes the fuel may be considered as consisting of carbon and hydrogen, although it must be pointed out that as the heat value of sulphur is extremely low (about 2219 lb. calories per pound) its presence should be allowed for if the quantity is at all abnormal.

The proportions in which the carbon and hydrogen are found vary somewhat according to the nature and origin of the original crude, or crudes, from which the fuel was distilled, but for fuels commonly available in this country the ratio quoted above may be taken as representative. The higher calorific value of such fuels varies from about 10,500 lb. calories to a little over 11,000 lb. calories per pound. The combustion of a fuel of the composition quoted will produce $0{\cdot}13 \times 9 = 1{\cdot}17$ lb. of water vapour, so that the lower calorific value will be $1{\cdot}17 \times 621 = 727$ lb. calories less than the figures just given, or, in round figures, 10,000 lb. calories per pound.

11. The Energy Content of the Cylinder Charge.

For purposes of simplifying engine calculations it is convenient to consider the energy content of the cylinder charge in terms of ft.-lb. per cubic inch. The engine measures its fresh charge as air only and not as a mixture of fuel and air, so that if the theoretically correct mixture of fuel and air is used we shall be adding the energy contained in 1 lb. of fuel to that already contained in 14·514 lb. of air. Under the standard conditions of pressure and temperature the energy supplied to 1 c. in. of air will be

$$\frac{10{,}000 \times 1400}{14{\cdot}514 \times 12{\cdot}4 \times 1728} = 45 \text{ ft.-lb. per standard cubic inch.}$$

If the quantity of fuel supplied is insufficient to utilize the whole of the available oxygen, the energy content will be reduced in direct proportion to the fraction of the air which is actually used. The figure of 45 ft.-lb. may be taken as representing the energy available per standard cubic inch of air for any mineral hydrocarbon that is likely to be utilized as a fuel for any internal-combustion engine, departures from this figure being of the order of not more than 2 per cent.

The conditions under which combustion takes place in compression-ignition engines are such that it is not found possible to utilize the whole of the available oxygen, about 85 per cent being the maximum which can be utilized efficiently. The maximum quantity of energy which can be supplied to the engine under normal conditions is there-

fore reduced to $0{\cdot}85 \times 45 = 38{\cdot}25$ ft.-lb. per standard cubic inch. It may be mentioned in passing that not many engines attain to a figure of 85 per cent, a much lower figure being quite usual. The type of combustion chamber has a direct bearing upon the use which can be made of the air, those having a high swirl velocity enabling a greater proportion of air to be used than those having a low swirl velocity. Under practical conditions the air used varies from as low as 60 per cent up to the figure quoted above, although a figure of 75 to 80 per cent is not very often exceeded.

Under service conditions the air received by the engine is hardly ever under standard conditions of temperature and pressure, but at a density somewhat less than the standard. The energy which can be supplied to 1 c. in. of engine cylinder volume is therefore reduced correspondingly. The maximum absolute volumetric efficiency (p. 88) of an engine under full load is around 80 per cent, so that the maximum quantity of energy which can be supplied to 1 c. in. of cylinder volume will be $38{\cdot}25 \times 0{\cdot}80 = 30{\cdot}6$ ft.-lb./c. in.

Making allowance for the fact that neither the maximum quantity of the air which can be used nor the maximum absolute volumetric efficiency are hard-and-fast values, it will be convenient to take the maximum quantity of energy which can normally be released per cubic inch of cylinder volume as 30 ft.-lb. Taking the more conservative figure of 75 per cent for the volumetric efficiency and 80 per cent for the maximum air used, we get a figure of 27 ft.-lb. per cubic inch cylinder volume as a fair practical figure.

12. Power and Mean Effective Pressures.

Once the energy released per cubic inch of cylinder capacity is known, the power and the mean effective pressure which will be developed are readily calculated.

To ascertain the power which will be developed from a given energy content it is only necessary to multiply the energy figure in ft.-lb. per cubic inch by the number of cubic inches supplied per minute and divide by the usual H.P. constant of 33,000:

$$\frac{\text{ft.-lb./c. in.} \times \text{c. in./min.}}{33{,}000} = \text{H.P.}$$

Similarly, to determine the mean effective pressure it is only necessary to multiply the energy content in ft.-lb./c. in. by 12:

$$\frac{\text{ft.-lb.}}{\text{c. in.}} \times 12 = \frac{\text{lb.}}{\text{sq. in.}}$$

The above figures are, of course, on the basis of 100 per cent conversion. The values for any efficiency less than 100 per cent will be

obtained by multiplying the equation by the efficiency of the conversion, i.e. by the indicated thermal efficiency of the engine if it is the indicated H.P. or M.E.P. that is required, or by the brake thermal efficiency if the answer is required on a B.H.P. basis. If the efficiency value taken allows only for the losses sustained from the several different causes, the result obtained must be further multiplied by the gain obtained from the increase in volume of the gases which follows combustion, the M.E.P. and therefore the H.P. being increased in proportion to the increase in volume of the gases.

If we know the energy to be supplied per cubic inch of cylinder volume and the efficiency of the cycle upon which the engine is intended to work, it is easy and simple to determine the output obtainable from any given engine. With an energy content equivalent to the maximum attainable under standard conditions and using the chemically correct mixture, the M.E.P. developed at 100 per cent efficiency would amount to as much as 540 lb./sq. in., while with the figure of 30 ft.-lb./c. in. the M.E.P. works out at 360 lb./sq. in. If allowance is made for the losses occasioned by the physical properties of the gases themselves (Chap. IV), but the assumption is made that no other losses take place, the M.E.P.s produced when different values of expansion ratios are employed are those shown in Table III.

TABLE III

Expansion Ratio	Theoretical Efficiency at 85% air used	I.M.E P lb./sq. in. at 30 ft.-lb./c. in.
10	47·5	171
12	50·1	180
14	52·3	188
16	54·2	195
18	55·8	201

13. Air Consumption.

These deductions bring us to a point which it is essential to keep in mind when dealing with problems connected with all forms of internal-combustion engine: the output obtainable from a given engine depends not upon the quantity of fuel which can be passed through it in a given time, but upon the *quantity of air it can be made to handle and the use that can be made of this air.*

In the case of a liquid fuel engine there is no difficulty in getting any desired quantity of fuel into the engine, but there is nothing to be gained by supplying it with any more fuel than can be consumed economically by the air it has received. In making this general statement, however, the fact must not be overlooked that the maximum output from a petrol engine is obtained when using a mixture con-

taining an excess of fuel. The performance of an engine is commonly judged upon the fuel it consumes while doing a unit of work. This is only natural, because the fuel has to be purchased whereas the air consumed is obtained free. It is only right that the financial side of power production should receive due attention, but the fact that the fuel has to be purchased has focused attention upon the fuel, whereas it is from the use made of the air that the all-round efficiency of an engine can best be judged and compared with that of other engines. This is particularly true of the compression-ignition engine, which, owing to the way in which the fuel is introduced, is incapable of making use of the whole of the air it receives.

Ricardo has stated that if we express the results obtained from an internal-combustion engine in terms of pounds of air per H.P. hour we have a much better basis of comparison than when the fuel is used. This method would not at first sight appear to give satisfactory results when applied to the compression-ignition engine, because the quantity of air per cycle received by the engine remains substantially constant regardless of load. Working on this basis, we get a rapidly increasing weight of air per H.P. hour as the load is decreased, suggesting a serious falling-off in efficiency instead of exactly the opposite tendency. Unlike the fuel, however, any air supplied to the engine in excess of that which is consumed is not really wasted. After passing out through the exhaust it is returned to the atmosphere and is available for use at some future time. We are therefore justified in expressing our results upon a basis of the weight of air actually burned per H.P. hour, and the method thus becomes applicable to the compression-ignition engine as well as to the carburetting engine.

To ascertain the weight of air actually used per hour it is not necessary to make a measurement of the air consumption. All that is necessary to obtain the weight of air burned per H.P. hour is to multiply the weight of fuel supplied to the engine by the weight of air required for the combustion of 1 lb. of fuel, 14·514, say 14·5, and divide the result by the H.P. developed.

The air which has actually been consumed will have resulted in the liberation of 45 ft.-lb. of energy per standard cubic inch. The volume of air required per H.P. hour may therefore be determined from the equation given on p. 37 for determining the H.P.:

$$V = \frac{33{,}000 \times 60}{45E} = \frac{44{,}000 \text{ c. in./hr.}}{E} = \frac{25{\cdot}5}{E} \text{ c. ft./hr.},$$

where E is the efficiency of the engine.

The weight of air required to develop one H.P. will therefore be

$$W = \frac{25{\cdot}5}{12{\cdot}4E} = \frac{2{\cdot}06}{E} \text{ lb./hr.}$$

Under actual operating conditions 30 ft.-lb. is the maximum amount of energy that can be liberated per cubic inch of cylinder volume, and the cylinder capacity necessary to develop one H.P. will therefore be increased proportionately, the actual volume being obtained by substituting the figure 30 for 45 ft.-lb. in the above expression. Thus

$$\text{Vol. swept per H.P.} = \frac{33{,}000 \times 60}{30E} = \frac{66{,}000}{E} \text{ c. in./hr.} = \frac{38{\cdot}2}{E} \text{ c. ft./hr.}$$

The volumes and weights of air theoretically required after allowance has been made for the losses occasioned by the physical properties of the gases themselves, but assuming that no other losses are sustained, are given in Table IV for expansion ratios between 10 and 18.

TABLE IV

Expansion Ratio	Theoretical Eff % at 85% air used	Weight of Air lb./H.P. hr.		Vol. of Air used at N.T.P. c. ft./H.P. hr.	Vol. swept by Piston c. ft./hr.
		100% used	85% used		
10	47·5	4·34	5·10	53·82	80·4
12	50·1	4·11	4·84	50·96	76·4
14	52·3	3·94	4·64	48·85	73·04
16	54·2	3·80	4·47	47·12	70·5
18	55·8	3·69	4·33	45·63	68·5

The weight of air actually *used* varies in practice from about 5·25 to about 6·25 lb. per B.H.P. hour at powers in the neighbourhood of the maximum output, while the corresponding figures upon the I.H.P. basis vary from about 4·0 to 4·5 lb. per I.H.P. hour.

The weight of air *supplied* to the engine per B.H.P. hour at the maximum output is a very useful guide to the all-round efficiency of the engine and clearly indicates the scope for further improvement in output. The figure normally obtained runs from about 8 to about 10 lb. air per B.H.P. hour. An engine which has a high thermodynamic efficiency is not necessarily one making good use of the air it receives, and consideration of the performance of an engine solely upon a basis of fuel consumption does not give the whole story. As a matter of fact, it will frequently be found that engines which have a high efficiency upon a fuel basis have a relatively poor efficiency from the point of view of their ability to make use of the air they receive. To quote an example: two different forms of combustion chamber on the same experimental engine at the same speed gave almost exactly the same B.M.E.P. at the limit of clear exhaust, but one had a fuel consumption of 0·387 lb. per B.H.P. hour and the other a consumption of 0·42 lb. per B.H.P. hour. There is no question which

is the better of the two from a thermodynamic standpoint, but the engine with the higher consumption was actually burning a greater quantity of air than the other one in the ratio of $\cdot 42/\cdot 387 = 1\cdot 086$. If, therefore, the engine with the better consumption could be made to utilize its air as effectively as the other, then the B.M.E.P. would be increased in like proportion, and instead of the figure of 101 lb./sq. in. which was actually obtained a figure of 110 lb./sq. in. would have been reached. In this particular instance there was little difference between the indicated mean pressures developed, the difference in the B.M.E.P. being due to energy absorbed in producing a compression-induced swirl. A little modification of the combustion chamber of the swirl-chamber engine resulted in a small improvement in consumption accompanied by a material increase in B.M.E.P. at the limit of a clear exhaust, and a much greater output was obtained, although still at the expense of some increase in fuel consumption as compared with the other engine.

For development and research work, therefore, it is not enough to know how many pounds of air are being burned in the cylinder in a given time, but we must know also what proportion the air burned bears to the total weight of air handled by the engine. Unless we know this we cannot tell whether a failure to develop the required M.E.P. is due to the inability of the engine to make use of air which is actually available, or to a shortage of air from a lack of breathing capacity. For this class of work, therefore, some means for measuring the air or some other suitable method for determining the use made of the air is an essential part of the experimental equipment if a high output is to be obtained with a minimum of wasted effort.

When means are available for determining the proportion of the air which is used, the best method of comparing results is to plot them on the basis of the air used. The air used, expressed as a percentage of the air available, is a non-dimensional factor and enables engines of widely differing sizes to be directly compared. The specific air and fuel consumptions, B.M.E.P., I.M.E.P., efficiency on the brake and indicated horse power, exhaust temperatures, &c., can all be plotted on this basis.

In cases where means are not available for determining the proportion of air utilized, a convenient method of plotting results is on the basis of the energy supplied per unit volume. This figure embraces the air used and the volumetric efficiency, and is readily determined from the readings normally taken during every bench test. From the point of view of the results obtained from an engine it makes little difference whether the maximum M.E.P. is limited by a failure to utilize air which is available, or a failure to get the air into the cylinder.

Curves of engine performance plotted on the basis of the energy supplied or of the air used are essentially the same in form. The real difference lies in the scale of the base line, and if desired the base line

may be arranged to show both the air used and the energy per unit volume. It should be pointed out, however, that a slight distortion of the scale may result from the small change in volumetric efficiency which usually takes place with a change in load. One great advantage

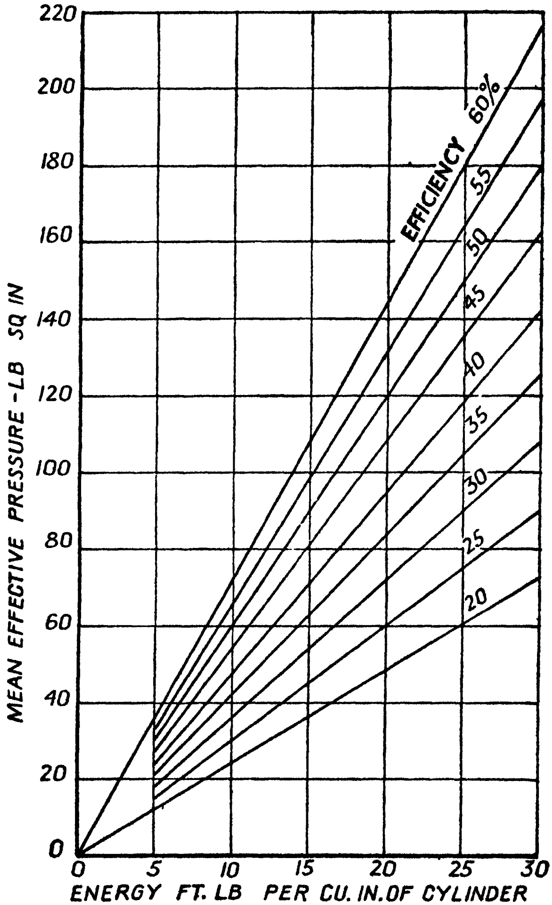


Fig. 17.—Relationship between energy, efficiency and mean effective pressure

of these two bases of comparison is that if a standard scale is adopted for all factors, the results obtained from all engines are directly comparable regardless of size, and once we have become accustomed to the form of the diagrams, the merits of a new set of readings are very quickly apprised without the necessity for direct comparison with another set of readings.

At a constant value of efficiency a curve of M.E.P. plotted against the air used, or energy per unit volume, will be a straight line, the slope of the line being a measure of the efficiency. Fig. 17 shows the M.E.P. which will be developed at different values of efficiency for energy values up to 30 ft.-lb./c. in. of cylinder capacity and, as will be seen, consists of a series of straight lines radiating from the common zero. Such a diagram is useful for comparing results obtained from engine tests, because readings plotted in this way show how the M.E.P. varies with the energy supplied and at once shows up any sudden decrease which may occur in the rate of increase of M.E.P. It also gives an indication of the scope for improvement.

When readings of B.M.E.P. are plotted in this manner, zero M.E.P. and zero energy do not of course coincide, because of the energy necessary for overcoming engine friction and pumping losses. The energy required at zero B.M.E.P. will depend upon the cycle efficiency as well as the mechanical losses, and the slope of the diagram at this point will enable both to be obtained with a reasonable degree of accuracy, as is described later.

CHAPTER IV

The Losses and Limitations of the Practical Engine

1. The Influence of the Change in Specific Heat.

The increase with temperature of the specific heats of gases has an important influence upon all forms of the internal-combustion engine. The direct effect of the increase in the specific heat is that the increase in the temperature of a gas is not directly proportional to the quantity of heat added to it, and the temperature therefore increases at a lower rate than the quantity of heat contained in the gas. Also, the increase in temperature resulting from the addition of a given quantity of heat to a given weight of gas decreases as the initial temperature of the gas is raised.

When a gas is heated under constant volume conditions the pressure of the gas after being heated is given by the product of its initial pressure and the ratio of its absolute temperatures before and after being heated. It follows, therefore, that if the specific heat increases with temperature, the ratio of the pressures before and after being heated will be less than the ratio of the quantities of heat contained in the gas before and after being heated. That is:

$$\frac{P_2}{P_1} < \frac{H_2}{H_1},$$

and in effect the pressure rise is less than that which would have been produced had the specific heat remained constant.

As it is solely by the changes in pressure produced by changes in the heat contained in the gas that the engine is able to convert heat energy into mechanical work, any reduction in the pressure increase from a given addition of heat means less ability to do work and therefore a loss in efficiency.

As has already been stated, a gas has two specific heats, one when it is heated at constant volume and the other when it is heated at constant pressure, the difference between the two being the work done in pushing back the surrounding atmosphere when the heating is done

at constant pressure. We can therefore express the specific heat at constant pressure in terms of that at constant volume, thus:

$$K_p = K_v + a,$$

where a is a constant for all gases and represents the work done against the surrounding atmosphere.

It has also been shown that the efficiency of all internal-combustion engines is governed primarily by two factors, the expansion ratio employed and the ratio of the two specific heats, γ:

$$\gamma = \frac{K_p}{K_v} = \frac{K_v + a}{K_v} = 1 + \frac{a}{K_v}.$$

From this expression it is clear that the ratio of the two specific heats must change as the specific heats increase, because the increase is due to an increase in the specific heat at constant volume, K_v, only.

By applying the modified expression for γ to the equation for the efficiency of the several idealized cycles of operation we shall see at once how the change in K_v affects the efficiency. For present purposes it will be sufficient to examine the equation for the constant volume cycle, since the expression for this cycle forms a basic part of the expressions for the other two cycles.

The expression for the efficiency of the constant volume cycle is

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1},$$

and by substituting for γ the expression just derived, we get

$$E = 1 - \left(\frac{1}{r}\right)^{a/K_v}.$$

From this it will be clear that if a remains constant the value of $\left(\frac{1}{r}\right)^{a/K_v}$ must increase as K_v increases and there will be a corresponding decrease in E. As the value of K_v increases with the temperature, it follows that *the efficiency of the cycle must decrease as the maximum temperature increases.*

2. Dissociation.

Very closely allied with the change in specific heat in its influence upon engine performance is the phenomenon known as *dissociation.* Many chemical reactions, among them those connected with combustion, are of a reversible nature and, given suitable conditions, may take place in either direction. Thus the reaction between hydrogen and oxygen to produce water vapour will, under suitable conditions, take place in the reverse direction and water vapour will break up into

its component elements of hydrogen and oxygen. Under this latter condition, however, the heat, or energy, which was evolved when the hydrogen and oxygen combined to produce the water must be replaced before the separation of the hydrogen from the oxygen can occur. The chemical equation for the reaction between hydrogen and oxygen may therefore be written

$$2H_2 + O_2 \rightleftharpoons 2H_2O + \text{Heat}.$$

Similarly with carbon dioxide; under certain conditions this gas will break up, not, however, into its constituent parts of carbon and oxygen, but into carbon monoxide and oxygen, provided heat is replaced equivalent to that liberated when carbon monoxide is burned to carbon dioxide. This reaction may similarly be represented thus:

$$2CO + O_2 \rightleftharpoons 2CO_2 + \text{Heat}.$$

The condition responsible for dissociation is high temperature, the extent of the dissociation increasing with the temperature. Under extremes of temperature the various forms of matter cannot exist in combination with each other, but exist only in an uncombined state, i.e. only as elements. The temperature at which a complete state of dissociation exists is far in excess of anything reached in an internal-combustion engine, but under certain conditions the temperatures reached are sufficiently great for an appreciable measure of dissociation to take place, notably when the conditions permit of the complete combustion of the whole of the available oxygen.

What takes place in an internal-combustion engine might more correctly be described as non-association, or delayed association, rather than actual dissociation. It is not so much the case of an existing substance being resolved by a high temperature into its constituent elements, as of the elements being unable to complete their reaction on account of the temperature they themselves have produced by their interaction. The reaction therefore ceases, or at least slows down, until a loss of heat reduces the temperature and allows the process to continue. It is of course quite possible that combinations already formed may be broken up again by the heat produced by later combinations, and a circle of association, dissociation and reassociation may be formed during the height of combustion. A state of equilibrium will be reached in which combustion will proceed comparatively slowly and will thus be prolonged into the expansion stroke.

The full benefit of the expansion stroke cannot be derived from any heat which is liberated after the stroke has started, and the efficiency of the cycle suffers in consequence. The general effect of dissociation is a suppression of a part of the heat during the combustion period and the liberation of it as expansion proceeds, a condition which is really identical with the effects produced by the change in specific

heat. On account of this similarity, and also because at the higher temperatures it is practically impossible to separate these two influences, it is usual to take the two conjointly and treat them as if they were a single phenomenon.

3. The Effect of Changing Specific Heat and Dissociation Shown Graphically.

The effect of the change in specific heat and of dissociation may perhaps be more readily followed if shown graphically as in fig. 18, which illustrates their effect upon the constant volume cycle. The ideal diagram is shown by the full line diagram ABCD. Starting at A,

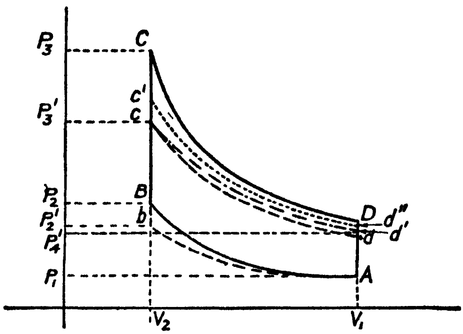


Fig. 18

where the pressure is P_1 and the volume V_1, the gas is compressed to volume V_2. Owing to the specific heat increasing with the temperature, the compression curve will not follow the true adiabatic curve AB to reach a pressure P_2 at volume V_2 but will follow some lower curve Ab to reach a lower pressure P_2'. The mean specific heat being greater than that which prevailed at A, the energy added to the gas during compression is insufficient to raise the temperature high enough to bring the pressure at volume V_2 up to P_2, but only to the lesser value P_2'.

If now we add the same quantity of heat as would under ideal conditions have raised the pressure from P_2 to P_3, we shall obviously fall short of P_3 because we are starting from a pressure P_2' which is less than P_2. The specific heat will be increased yet further by the addition of more heat, so that the increase in temperature $T_3 - T_2$ of the theoretical diagram will not be realized and the final pressure at C will be reduced to P_3', as shown at c, instead of P_3.

Expansion now starts from c at pressure P_3', and if the value of γ were at its ideal figure of 1·4 the pressure during expansion would

follow the curve *cd*. The effect produced by the specific heat decreasing in value as the temperature falls is the same as if heat were being added progressively throughout the expansion stroke. The expansion curve will lie above the adiabatic curve *cd* as shown by *cd'*, and a final pressure P_4' will be reached at volume V_1. Heat is now rejected and the pressure again returns to P_1 at volume V_1, point A.

The work done is represented by the area *Abcd'*, and is manifestly less than the area ABCD, representing the ideal cycle. One point requires some further explanation. It might appear at first sight as if the change in specific heat had a beneficial effect during expansion. The expansion curve *cd'*, which takes into account the change in specific heat, lies above that which would be followed had the specific heat

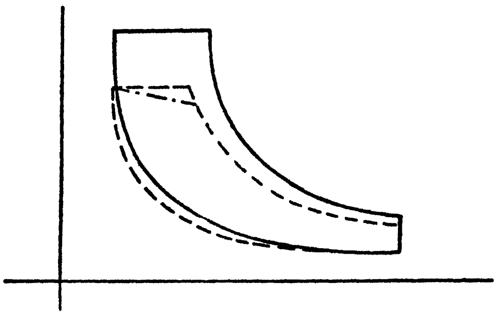


Fig. 19

remained constant, with the result that the area of the diagram is increased by the area *cdd'*. If, however, the specific heat had remained unchanged after the point *b* was reached, the point *c* would have been raised to *c'*, and expansion would have followed the line *c'd''*, with the result that an area *cc'd''d'* would have been added to the diagram. The area *cdd'* represents the work done by the reappearance of the heat which has been "suppressed" and does therefore represent some recuperation, but if no "suppression" had taken place a considerably larger area *cc'd''d'* would have been added to the diagram.

The effect of the change in specific heat is precisely similar when the constant pressure cycle is used. This is illustrated diagrammatically in fig. 19, which can readily be followed from the reasoning just given for the constant volume cycle. One point, however, which may not perhaps be immediately apparent is that in order to maintain the pressure constant the heat will have to be added at a higher rate than would have been necessary if there had been no change in specific heat. The expansion ratio which occurs during heat reception will therefore be reduced somewhat; this will tend to diminish, if only slightly, the loss which the change in specific heat entails.

4. The Efficiency Theoretically Attainable after allowing for the Properties of the Working Fluid.

In arriving at the equation for determining the efficiency in accordance with the air standard, the working fluid was assumed to have certain properties which it does not in fact possess, and as a result the values given by this equation are far in excess of anything which is at all possible of attainment. This has led to doubts being expressed as to the utility of the air standard as a practical basis of comparison. The question was discussed at some length by Tizard and Pye in the Empire Motor Fuels Report,* where they sum up the objects of a reference standard as follows: "There are different senses in which the 'standard engine of comparison' may be looked upon as the ideal with which real and imperfect engines may be compared. There is an essential difference, for instance, between an engine of comparison which postulates a working substance such as 'ideal air', and one which takes account of the properties of the real working substance and only defines and simplifies the working conditions by assuming, for instance, that the heating of the working substance is instantaneous, and that there shall be no loss or gain of heat by conduction through the cylinder walls. The first, from the nature of things, must ever remain far removed from practical possibilities, while the second, by taking into account the fundamental physical conditions and properties of the substances involved in the working of an actual engine, becomes an approachable, though not, so far as can be seen, an attainable ideal."

The "physical conditions and properties" referred to are, of course, the increase in the specific heats as the temperature increases and dissociation, and the efficiency which could be realized after taking these two items into account is discussed at some length in the report from the point of view of the petrol engine. Apart from their volatility, the hydrocarbon fuels used for petrol engines do not differ materially from those used for compression-ignition engines, and deductions made in respect of the former may reasonably be applied to the latter where the essential factors are the same for each. It was shown by Tizard and Pye that the general form of the equation for the efficiency of the cycle remains the same as for the air standard, the only adjustment necessary being to the value of the exponent. This for the air standard is taken as being 1·4, the ratio of the specific heats of pure dry air at ordinary temperatures, and, to make allowance for the change in specific heat and dissociation, a figure must be used which will represent the apparent value of the ratio of the two specific heats (plus an allowance for any dissociation which may

* *Proc. Inst. Auto. Eng.*, Vol. XVIII, Part 1.

occur) over the range of temperatures involved. This apparent value is usually designated by the symbol n, and for the chemically correct mixture this figure is given as 1·258, while for a mixture 20 per cent weak, i.e. one which contains sufficient fuel to utilize only 80 per cent of the available oxygen, the figure is given as 1·296, and increased further to 1·33 when the mixture is 50 per cent weak.

This increase in the value of n as the mixture strength is decreased is a point which has a very important influence upon the efficiency of the compression-ignition engine under its average operating conditions relative to that of a petrol engine under the same conditions. From the commercial standpoint the efficiency of any prime mover is judged, not by its performance under any one set of conditions, but by its average efficiency attained under the varying conditions of every-day service, and it is under these conditions that the compression-ignition engine shows to special advantage compared with the petrol engine.

On account of the narrow range of mixture strengths which have a suitable degree of inflammability, the power of the petrol engine is regulated to meet the demands upon it by the use of what is termed "quantity control" or "throttle control". That is, in order to meet the varying demands for power the quantities of fuel and air received by the engine are varied in unison. The ratio fuel/air remains substantially constant and, subject to the influence of the residual exhaust gases, the relative volume of which will increase as the volume of the fresh charge is reduced, the quantity of heat supplied per unit weight of charge is constant under all conditions of load. The maximum temperature reached therefore remains substantially constant, and the ratio of the apparent specific heats will therefore remain the same at all loads. The efficiency of the cycle is thus unaltered by changes in load.

On the other hand, the power developed by the compression-ignition engine is governed by "quality control" or "mixture control". The engine draws in a fixed charge of air under all conditions of load and the weight of the fuel charge alone is adjusted to meet the demands upon the engine. The quantity of heat added per unit weight of air thus decreases as the load decreases; the maximum temperature is therefore reduced also, and with it the specific heats. Thus *the efficiency of the cycle increases as the load carried by the engine decreases*, and provided that the combustion efficiency remains unimpaired, will continue to increase right down to the smallest possible quantity of fuel which can be injected. The trend of a load-efficiency curve is towards the air standard figure at zero load. Actually, however, the air standard figure will not be reached if the curve is extrapolated to zero, because the compression temperature, which at zero fuel becomes the maximum temperature, is itself sufficiently great to produce some increase in the specific heats.

The increase in the value of n with a reduction in mixture strength as given by Tizard and Pye therefore has a special significance for the compression-ignition engine. The figure quoted for the chemically correct mixture has no practical application for these engines, because it has not yet been found possible to utilize the whole of the available oxygen, about 85 per cent being about the absolute maximum that can be used. The figure $n = 1{\cdot}296$ for a 20 per cent weak mixture may, however, be considered as applicable, as may also that of 1·33 given for 50 per cent air used.

From the foregoing it will be evident that when we are dealing with the compression-ignition engine and wish to make allowance for the physical properties of the actual working medium we do not, as is the case with the air standard, have for the efficiency of our engine a single value which holds good for all conditions of operation and depends solely upon the expansion ratio, but a changing value, depending primarily upon the expansion ratio, but varying inversely with the maximum temperature. In order, therefore to determine the efficiency of an engine relative to the efficiency theoretically attainable if allowance is made for the properties of the working medium, we must first of all determine how the standard varies and then choose the figure corresponding to the conditions under which the engine is working. To do this it is necessary to have some definite basis on which to work. This is readily provided if we compare all our results on the basis of the percentage of the available air which has been used up in consuming the fuel charge received by the engine. In so doing we are indirectly comparing the results in terms of the maximum temperature. The various hydrocarbon fuels differ only slightly in their composition and the quantity of heat liberated when they are burned with the theoretically correct weight of air is very nearly the same for all. To speak of burning a given proportion of the air supplied is therefore equivalent to saying that a certain amount of heat has been added per unit weight of air, i.e. its temperature has been raised by a given amount. The temperature of the air before the heat is added will cause some variation in the final temperature, but in actual practice the compression temperature does not vary over very wide limits, certainly not wide enough to cause any serious discrepancy.

The specific heats of gases do not all increase at the same rate. Of the four gases, O_2, N_2, CO_2 and H_2O, which, mixed together in varying proportions, form the working medium of the compression-ignition engine, the specific heats of CO_2 and H_2O increase at a greater rate than those of O_2 and N_2, and the former gases are subject to dissociation at high temperatures. The CO_2 and H_2O are products of combustion, so that in an engine governed by quality control the specific heats of the mixture of gases will depend not only upon their temperature but also upon the composition of the mixture, i.e. it will vary with the

proportion of the air used in combustion. This will mean that at any given temperature the specific heats of the gases will increase, and the ratio of the two specific heats will decrease, as the proportion of the air used is increased. As the maximum temperature is governed by the proportion of air used, it is therefore necessary to take into consideration both the maximum temperature and also the composition of the gases before the value of n suitable to compression-ignition engine working can be obtained.

The accurate determination of the specific heats of gases presents considerable difficulties, and the results obtained by different investigators do not agree with one another as well as could be desired. One difficulty is to separate the effects of any dissociation from those produced by a change in specific heat, but as the effects of these two phenomena upon the performance of the engine are identical it is usual to treat the two conjointly and, instead of using the actual values of the specific heats, to work in terms of the total energy contained in the gas, or a mixture of gases, at the temperature under consideration. The temperatures are usually measured on the absolute (Centigrade) scale, while the energy is given in any convenient unit.

A very complete investigation into the specific heats of the gases involved in combustion processes was carried out by Goodenough and Felbeck.* No new experimental work was undertaken, but all the available data were collected and examined, and values of the specific heats up to 4000° abs. were deduced, together with an expression connecting the temperature and the total energy contained in the gases. Information relative to dissociation is given also.

Using the information given by Goodenough and Felbeck, but omitting, for reasons explained hereafter, any effects from dissociation, the energy content for temperatures up to 3500° abs. has been worked out under both constant volume and constant pressure conditions for the gases resulting from the utilization of a number of different fractions of the available air. The composition of the products of combustion was taken to be the same as is given in Table I, and allowance was made for the change in weight of the charge due to combustion. The results obtained are shown in figs. 20 and 21, fig. 20 referring to constant volume conditions and fig. 21 to constant pressure conditions.

Before the value of n appropriate to any given set of conditions can be obtained, it is necessary to know the maximum temperature reached under these conditions. To determine this we must first know the temperature of the gases at the moment combustion begins. This will be governed by the compression ratio, but under normal conditions does not vary over very wide limits, from about 850 to 900° abs. representing the usual range, and a figure of 875° abs. may therefore be taken as the average figure. To arrive at the final tem-

* Bulletin 139, Engineering Experiment Station, University of Illinois.

perature resulting from the addition of any given quantity of energy, the energy represented by the initial temperature of the gases must be added to the energy received from combustion. Up to the moment when combustion actually begins the charge may be considered as consisting of air only, and the energy contained by air at a temperature of 875° abs. may be taken as being 10 ft.-lb./std. c. in. for

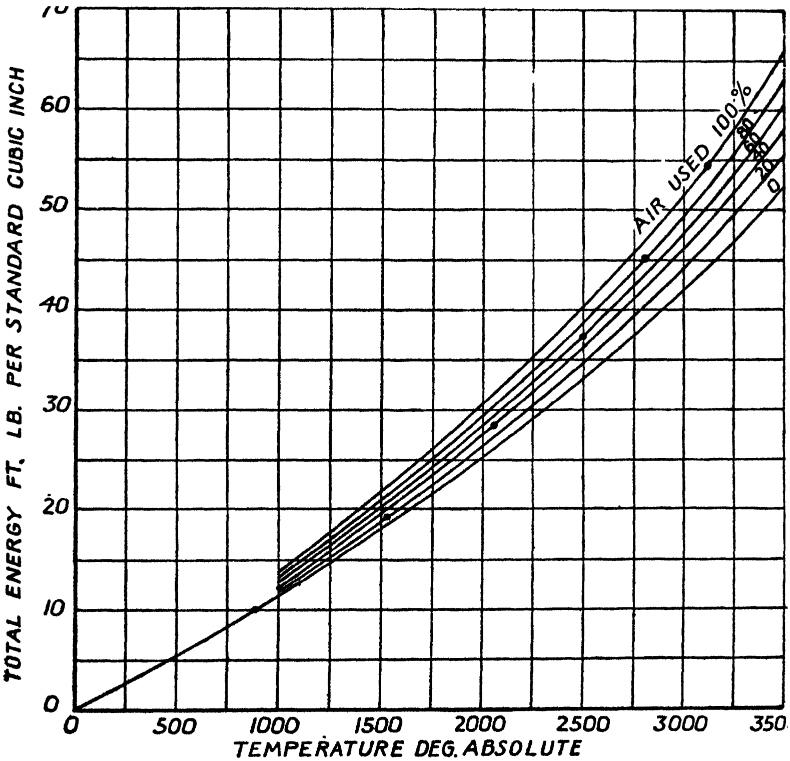


Fig. 20.—Temperature-energy curves for the gases resulting from differing degrees of air utilization at constant volume

constant volume conditions. This figure of 10 ft.-lb./std. c. in. must therefore be added to the energy received from the combustion of the air actually burned.

The combustion of a given fraction of air represents the addition of a definite quantity of energy to a mixture of gases having a definite composition, and the composition of the gases varies in a known manner (Table I, p. 24) with the quantity of air used. It follows, therefore, that, starting from air at a given temperature, the temperature after combustion must reach one of two temperatures only,

which of these two depending upon whether combustion takes place at constant volume or constant pressure. In order to arrive at this temperature we must know, therefore, not only the temperature from

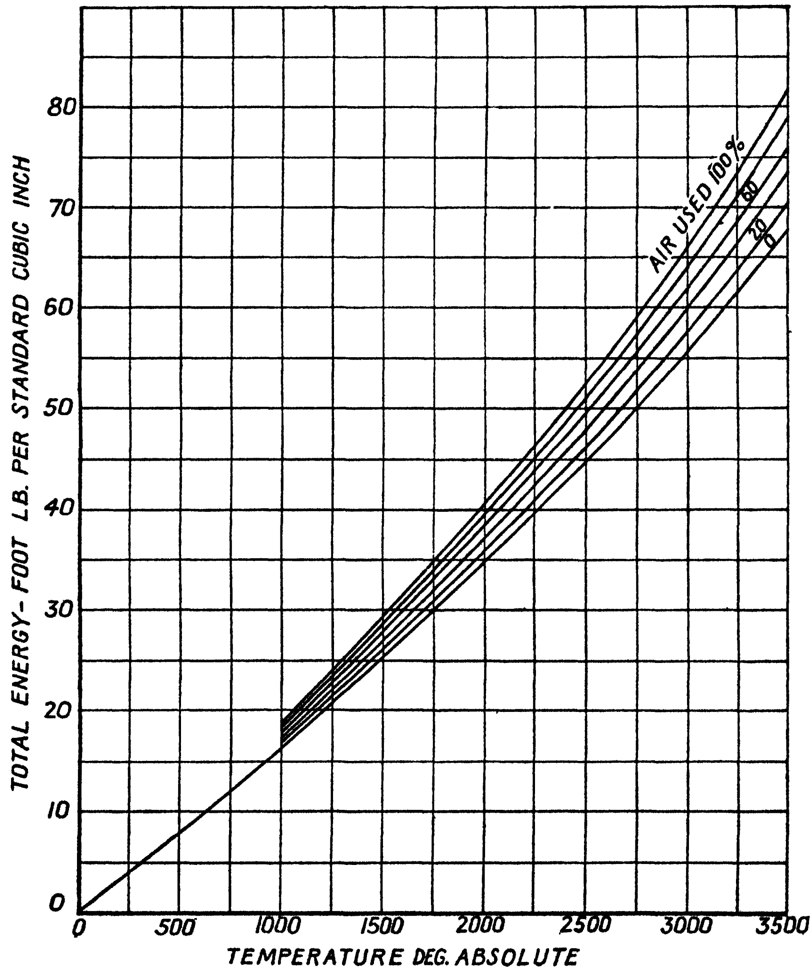


Fig. 21.—Temperature-energy curves for the gases resulting from differing degrees of air utilization at constant pressure

which combustion started, but also the temperature-energy curve corresponding to the quantity of air that is actually burned.

It has been shown (p. 36) that the complete combustion of the air results in the addition of energy to the extent of 45 ft.-lb./std. c. in., the energy varying in direct proportion when lesser quantities of air

are used. If, therefore, we start with air at 875° abs., energy 10 ft.-lb./std. c. in., and add energy progressively and without loss till the whole of the air is consumed, the energy content of the air will increase from 10 ft.-lb. up to the maximum possible figure of 55 ft.-lb./std. c. in. when the whole of the oxygen has been consumed. In increasing the energy from 10 to 55 ft.-lb., we have crossed over

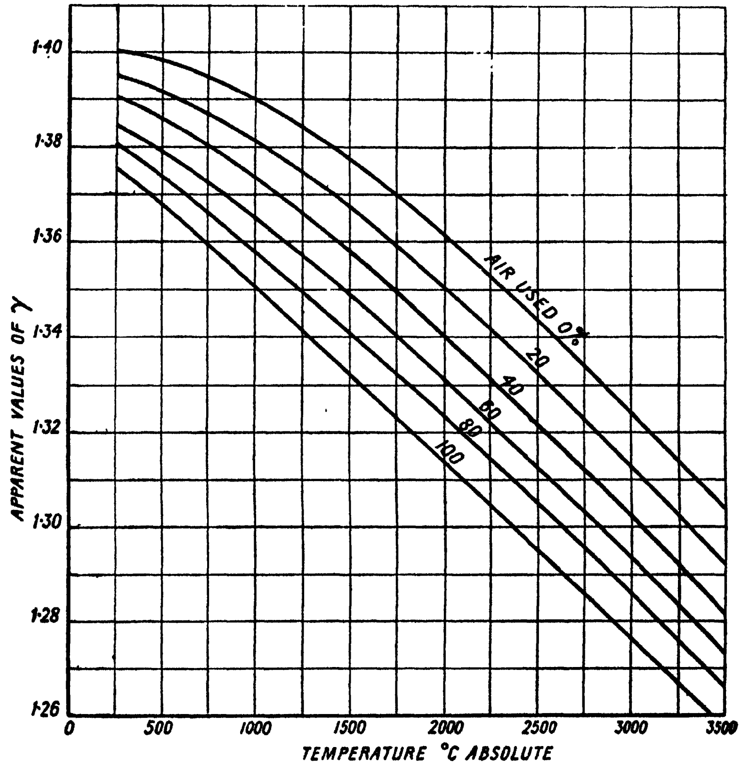


Fig. 22.—Apparent values of γ for the gases resulting from differing degrees of air utilization

from the point representing 10 ft.-lb./std. c. in. on the temperature-energy curve for pure air to the point representing 55 ft.-lb./std. c. in. on the 100 per cent air-used curve, as indicated by the points marked on each curve in fig. 20. The point at which the resulting diagonal curve intersects the curve for any intermediate value of air used will represent the temperature and energy corresponding to the utilization of that fraction of air. Thus, at 40 per cent air used we are adding $0{\cdot}4 \times 45 = 18$ ft.-lb./std. c. in., making the total quantity 28 ft.-lb./std. c. in., which corresponds to a temperature of 2055° abs.

By taking a number of intermediate points we may trace out the path followed in crossing from the zero air-used curve to the 100 per cent air-used curve, as shown in fig. 20, and the temperature reached by the combustion of any given fraction of the air may be obtained therefrom. The temperature being known, the value of n corresponding to the use of that fraction of air may be read from fig. 22. In order to make matters clear, Table V is included.

TABLE V

Air Used	Energy before Combustion	Ft.-lb /std c. in added by Combustion	Total	Temperature at Const. Vol. (abs)	n
0	10	0	10	875	1·394
20	10	9	19	1520	1·367
40	10	18	28	2055	1·338
60	10	27	37	2490	1·313
80	10	36	46	2830	1·290
100	10	45	55	3120	1·272

The values of n so obtained are shown plotted against "air used" in fig. 23, and from this may be obtained the value of n corresponding to the utilization of any desired fraction of air. By the use of these values for n, the efficiency obtainable after allowance has been made for the change in specific heat may be obtained by substituting for γ, in the air standard efficiency equation, the value of n appropriate to the conditions of operation. The values for efficiency so obtained for the range of expansion ratio normally used for high-speed engines are shown in fig. 24, which gives also the air standard values for comparative purposes.

It was mentioned that the effects of dissociation have been ignored in working out the foregoing results. This is justified on the following grounds. Dissociation takes place only at extremes of temperature, and while it exerts an appreciable influence at the temperatures reached when the whole of the available oxygen is burned, the fact that nothing like the whole of the oxygen can be burned in a compression-ignition engine results in much lower maximum temperatures and so limits the possible influence of dissociation. The maximum quantity of oxygen that can be burned even if no regard is paid to exhaust cleanliness is about 85 per cent, while about 80 per cent is as much as can be burned if a reasonably colour-free exhaust is maintained. Many engines do not normally use more than 60 to 65 per cent of the oxygen. Conditions assuming 100 per cent utilization of the oxygen therefore have no more than a purely academic interest. At 80 per cent utilization the maximum temperatures are reduced to a

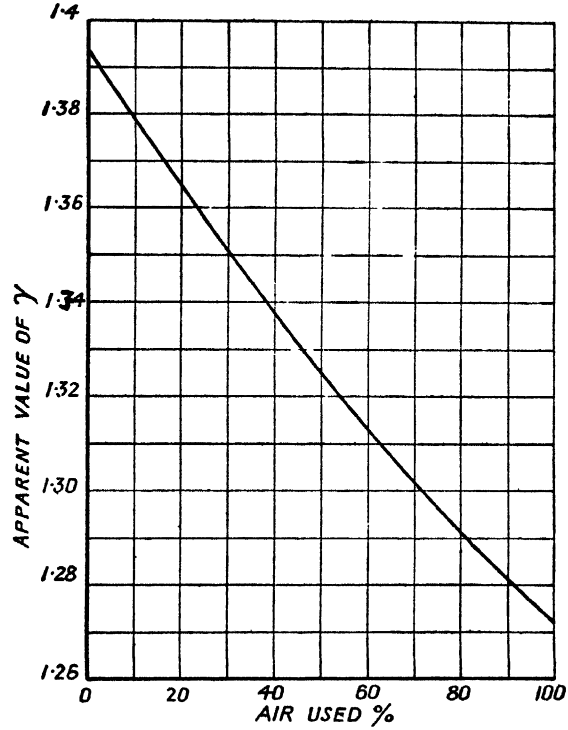


Fig. 23.—Apparent values of γ or n corresponding to differing degrees of air utilization

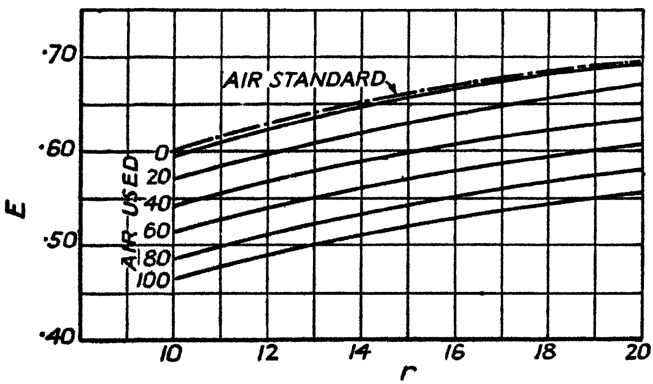


Fig. 24.—Efficiency attainable at differing degrees of air utilization when the change in specific heat is taken into account

figure at which, even under relatively low pressures, the possible amount of dissociation is much reduced. The influence of pressure upon dissociation provides a second reason for ignoring dissociation effects. The amount of dissociation decreases rapidly at high pressures, as will be seen from Tables VI and VII, which are quoted from Bjerrum.*

TABLE VI

Percentage Dissociation of CO_2

Temp. °C.	Pressure (Atmospheres)			
	0·1	1·0	10·0	100·0
1500	0·104	0·048	0·0224	0·01
2000	4·35	2·05	0·96	0·445
2500	33·5	17·6	8·63	4·09
3000	77·1	54·8	32·2	16·9

TABLE VII

Percentage Dissociation of H_2O

Temp. °C.	Pressure (Atmospheres)			
	0·1	1·0	10·0	100·0
1500	0·043	0·02	0·009	0·004
2000	1·25	0·58	0·27	0·125
2500	8·84	4·21	1·98	0·927
3000	28·4	14·4	7·04	3·33

In arriving at their values for *n* for conditions involving complete combustion Tizard and Pye were dealing with pressures of the order of 40 atmospheres reached after allowing for dissociation. The combustion of a compression-ignition engine begins under pressures of this order, and even with only 80 per cent utilization of the oxygen pressures of the order of 120 atmospheres will be produced under constant volume conditions. The temperatures reached at 80 per cent air used will be a trifle over 2500° C., so that the total dissociation will be something less than 5 per cent and at a very slightly smaller fraction of air utilization will be practically zero. At 100 per cent air used some dissociation would certainly take place, and though it would be appreciably less than under the conditions assumed by Tizard and Pye, it would have to be taken into consideration. Tizard and

* *Zeitschrift für Physikalische Chemie*, 1912.

Pye give the value of n as 1·258 for 100 per cent air utilization with dissociation; the figure found by the author for no dissociation is 1·272. The pressures produced with constant volume burning would be of the order of 130 to 140 atmospheres for 100 per cent air utilization, and at these pressures dissociation would be reduced to about two-thirds of that allowed for by Tizard and Pye, so that a value for n around 1·262 would appear to meet the case.

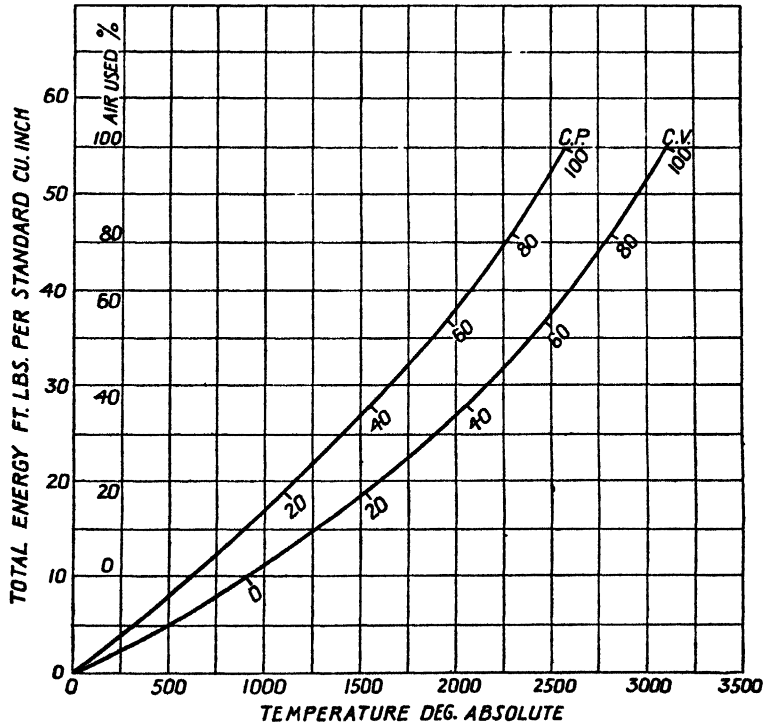


Fig. 25.—Temperature-energy curves for the gases resulting from the combustion of differing proportions of air at constant pressure and constant volume

For analysing engine cycles, diagrams such as figs. 21 and 22 with their multiplicity of curves are unnecessary; a single curve is all that is necessary to cater for each condition of burning. This curve will be the diagonal curve crossing the others from the zero air-used curve to the 100 per cent air-used curve. The small variation caused by differences in compression temperatures will be immaterial, because for a range of temperature of 850 to 900° abs. the change in energy will be only a little over 2 per cent either side of the mean figure of 10 ft.-lb./std. c. in. given above. The two diagonal curves for constant volume and constant pressure conditions are shown in fig. 25, which

gives both the total energy content and a scale for the amount of air used, and will enable an analysis to be made by either method at will.

The efficiency figures derived by the use of the value of n obtained in this way may be looked upon as a practical ideal standard, because they take into account the physical properties of the medium we are compelled to use. The figures shown in fig. 24 bring out very clearly the advantages of working upon quality control in the reduction of the losses occasioned by the increase in the specific heats. Within the normal range of working, i.e. from about 80 per cent air used down to 20 per cent, there is a gain in efficiency of about 17 per cent. This gain in efficiency is a very real advantage under practical conditions. Figures from the indicated efficiency follow very closely the trend given by the theoretical curve and show this progressive improvement in efficiency as the load is reduced. The brake thermal efficiency is of course influenced by the mechanical efficiency of the engine. At any given speed the mechanical losses remain approximately constant at all loads and the mechanical efficiency therefore decreases as the load is reduced. The brake thermal efficiency is obtained by multiplying together the indicated thermal efficiency and the mechanical efficiency, and the improvement in the thermal efficiency is sufficient to neutralize the fall in mechanical efficiency, so that the brake thermal efficiency is maintained at a substantially constant figure over a fairly wide range of loads. In cases where the air utilization factor is fairly high the improvement in indicated efficiency may be sufficient to produce an actual improvement in brake thermal efficiency at loads slightly less than the maximum load. The improvement in efficiency as the load is reduced is assisted by the earlier completion of combustion which occurs under practical conditions under light load conditions. With a high air utilization factor the prolongation of combustion due to the difficulty the air and fuel have of finding one another has a marked effect upon the efficiency under these conditions. This condition is associated with the necessity for limiting the maximum pressures, dealt with in the next section.

5. The Limitation to the Maximum Compression Ratio.

Unlike the petrol engine, the compression-ignition engine does not have imposed upon it by the characteristics of the fuel it uses any limitations as to the maximum compression ratio that can be employed. Actually the limitations set by the fuel are in the opposite direction; the ignition characteristics of the fuel limit the *minimum* compression that can be used. This, however, does not seriously concern us, because we are interested in using the maximum ratio that will give us any advantage in all-round operating efficiency, although if certain fuels, such as those containing a high percentage of aromatics, are to

be used successfully in high-speed engines, the minimum compression ratio does become a factor of importance.

If we take 80 per cent as representing the maximum quantity of the available air that can be burned economically, then the efficiency of the constant volume cycle, after making allowance for the change in specific heat, will be given by the expression $E = 1 - \left(\frac{1}{r}\right)^{0.29}$. The efficiency will increase with the value of r and, as fig. 24 indicates, a substantial improvement in efficiency will be gained by raising the value of r to a figure well above the average value now being used. The majority of engines built to-day have a compression ratio around 15 or 16 : 1, and at the latter figure the theoretical efficiency will be 55·05 per cent (fig. 24). Raising the ratio to 20 : 1 will increase the efficiency to 58·05 per cent, an improvement of over 5 per cent, while by raising the ratio to 25 : 1 an efficiency of 60·69 per cent is obtained, representing a gain of some 10 per cent over that given by a ratio of 16 : 1. Why, then, should advantage not be taken of this improvement which ~~theoretical considerations say is possible? The answer is that~~ mechanical considerations necessitate that a limitation be placed upon the maximum pressures which are employed.

The maximum pressure reached at the end of combustion will depend upon the amount of heat added and the temperature and pressure at which combustion starts; these latter depend upon the compression ratio and the conditions existing when compression begins. Under average conditions, as is shown in the next chapter (p. 90), compression begins with a temperature of about 340° abs., and from a pressure of around 13·5 lb./sq. in. abs. (if the compression is considered as starting from bottom dead centre), and the average value of n for compression may be taken as 1·35. The pressures and temperatures that are reached at the end of compression after starting from the above conditions are shown in fig. 26. The combustion of 80 per cent of the available oxygen results in the addition of 36 ft.-lb. of energy per standard cubic inch of air, and the pressures and temperatures resulting from the addition of this quantity of heat are included in fig. 27. The figures in the full-line curve are based upon the assumption that no heat loss takes place during combustion, while those in the chain-dotted line assume a loss of 6 per cent of the added heat during combustion. This diagram shows that at the higher values of r the maximum pressure produced by constant volume burning is exceedingly high, and unless some means are adopted for limiting the maximum pressure, loadings of an impossible magnitude will result.

The maximum pressure that can be tolerated is really a question of ability to carry the resulting total load without having to resort to inordinately heavy, and therefore costly, methods of construction. Furthermore, heavy moving parts impose limitations upon the

maximum speed at which the engine can be run and also serve to increase the mechanical losses. A somewhat higher maximum pressure is possible with a cylinder of small diameter than with a larger one,

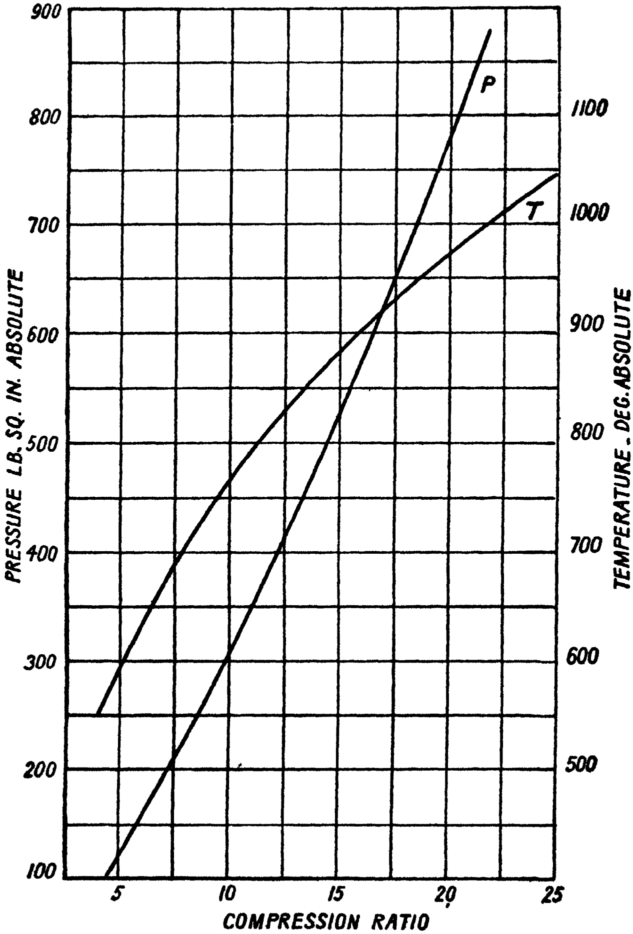


Fig. 26.—Absolute pressures and temperatures at the end of compression for compression ratios up to 25 : 1

because the scantlings of engines of small cylinder capacity are determined by foundry and manufacturing conditions rather than by stress considerations. This condition does not, of course, apply to bearing areas, which in all cases must be proportioned to the load they have to carry. As the cylinder size increases, however, the maximum load becomes more and more the governing factor in every

dimension, until a size is reached at which the total load becomes the sole criterion in design. Under such conditions the maximum pressure reached during combustion becomes a matter of first importance, because a small improvement in fuel economy may easily become

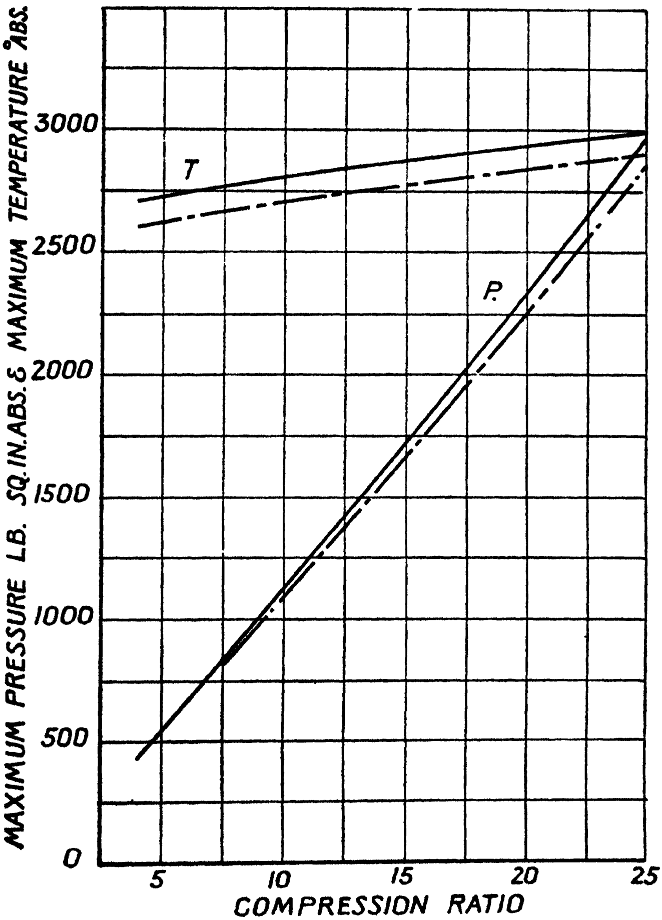


Fig. 27.—Curves showing the theoretical maximum pressures and temperatures with constant volume combustion when the change in specific heat is allowed for

uneconomical commercially if the weight and cost of the engine are increased beyond a certain figure.

Generally speaking, for engines having cylinders with a diameter around 4 to 5 in. and running at the higher speeds, a pressure of 900 to 1000 lb./sq. in. is as high as it is desirable to go, the lower figure

being preferred if conditions will permit. For engines running at the more moderate speeds and having cylinders of larger dimensions the limit is usually placed around 800 lb./sq. in.

If the maximum pressure is to be limited to some arbitrarily selected figure, then it follows that either the compression ratio must be limited to the figure which will permit all the fuel to be burned at constant volume without exceeding the desired maximum pressure, or if a ratio higher than this is chosen, the quantity of fuel burned at constant volume must be limited to that which will produce a pressure not greater than the desired maximum and the remainder of the fuel must then be burned at constant pressure. It will be seen from fig. 27 that if we are to burn the whole of the fuel at constant volume and keep within the limits of pressure just given, the compression ratio must not exceed 7 : 1 for a maximum pressure of 800 lb./sq. in. and 9 : 1 for a maximum pressure of 1000 lb./sq. in. if no heat is lost during combustion; if we allow for a loss of 6 per cent of the heat, then an additional half ratio can be given. At the other end of the scale, if the whole of the heat is burned at constant pressure, then the maximum pressure must not exceed the compression pressure and 800 lb./sq. in. will be reached with a compression ratio of 20·5 : 1 and 1000 lb./sq. in. will be reached with a ratio of 24·2 : 1. These two latter figures therefore represent the maximum possible compression ratio that can be employed when the maximum pressure is limited to the two figures given.

Between the limits of 7 : 1 and 20·5 : 1 for a maximum pressure of 800 lb./sq. in. and 9 : 1 and 24·2 : 1 for a maximum pressure of 1000 lb./sq. in. the fuel must be so divided between constant pressure and constant volume burning that the selected pressure is not exceeded, and the efficiency of the cycle will therefore vary between that obtained from the constant volume cycle at the lower ratio and that from the constant pressure cycle at the higher ratio. What we are interested to know, therefore, is how the efficiency will vary between these two limits and how far it will pay us to increase the ratio towards the upper limit.

It has been shown (p. 17) that the efficiency of the mixed constant volume-constant pressure cycle is given by the equation

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}\left[\frac{pr_e^{\gamma} - 1}{(p-1) + p\gamma(r_e - 1)}\right],$$

where r is the expansion ratio, p the ratio of the two pressures between which the constant volume burning takes place, r_e the ratio of the two volumes at the beginning and the end of the constant pressure burning, and γ the ratio of the two specific heats, equal to 1·29 for the mixture of gases produced by the combustion of 80 per cent of the

available oxygen and a mineral hydrocarbon fuel. The value of r_e cannot be conveniently determined by direct methods, but since $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$, and for expansion at constant pressure $P_2 = P_1$, we have $V_2/V_1 = T_2/T_1$. T_2/T_1, the ratio of the temperatures at the beginning and the end of the period of constant pressure expansion, can be determined from suitable temperature-energy diagrams, but for an analysis of this kind the chart in fig. 25 does not provide the necessary information because although what we are now concerned with is the combustion of 80 per cent of the available air, the combustion is divided between constant volume and constant pressure burning in proportions which will vary with the compression ratio in such a way that a given maximum pressure will not be exceeded. A further complication is that combustion will start from a different temperature for each ratio of compression, so that the total energy at the end of combustion, and therefore the maximum temperature, will vary with the compression ratio. The diagram required will include the temperature-energy curve for the utilization of 80 per cent of the air at constant pressure, as shown in fig. 21, and a series of curves similar to the diagonal indicated by the points marked on the temperature-energy curves shown in fig. 20 and reproduced as the constant volume curve in fig. 25. Actually a different diagonal curve will be required for every different temperature from which combustion may begin, but as the extremes of the temperature range are not very wide, from about 700 to 1000° absolute being the extreme limits, and the slope of the diagonal curve does not differ greatly from that of the others, a single curve can be drawn which will be accurate enough for all ordinary purposes. For this the constant volume temperature-energy curve from fig. 25 may be used, and fig. 28 shows this curve and the temperature-energy curve for the combustion of 80 per cent of the air at constant pressure combined to form the diagram by means of which the mixed cycle may be analysed.

To determine the efficiency of the mixed cycle the procedure is as follows: first determine the compression pressure and temperature for the required compression ratio. This will give both the energy contained by the air at the end of compression, from fig. 28, and p, the ratio of the compression pressure to the maximum pressure selected. The absolute temperature at the end of the constant volume burning is obtained by multiplying the compression temperature by the ratio of pressure rise. The temperature at the end of the constant volume burning being known, the energy added during this period can be ascertained from the constant volume curve of fig. 28, and from the energy thus added the percentage of air burned may be ascertained on the basis that the addition of 45 ft.-lb./std. c. in. represents 100 per cent utilization of the air.

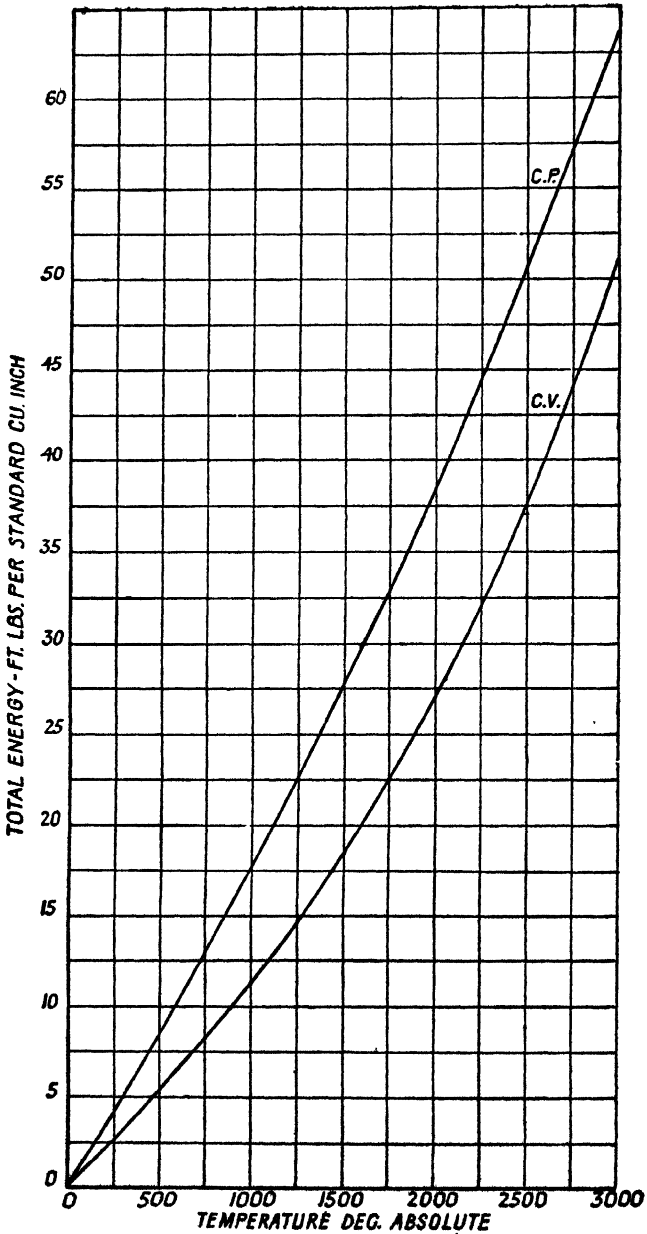


Fig. 28.—Chart for determining the efficiency of the mixed cycle when 80% of the available air is used

At the end of the constant volume burning we add additional heat without increasing the pressure or temperature to arrive at the constant pressure conditions, and we therefore transfer to the constant pressure line at the same temperature as that reached at the end of the constant volume burning. Further heat is now added at constant pressure until the whole of the heat representing 80 per cent utilization of the air, 36 ft.-lb./std. c. in., has been added. This gives the temperature at the end of the constant pressure burning, and the ratio

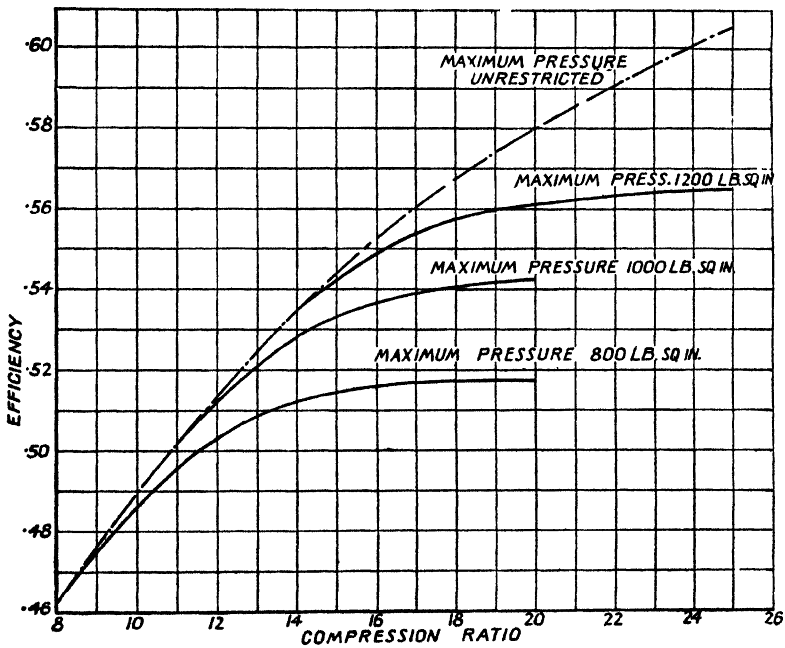


Fig. 29.—Effect upon efficiency produced by limiting the maximum pressure. Air utilized 80%; allowance made for change in specific heat

of this temperature to that at the beginning of the constant pressure burning gives the ratio of the volumes at the beginning and at the end of the constant volume burning, i.e. the value of r_e. We now have both p and r_e and are in a position to solve the efficiency equation.

With this diagram and the compression pressures and temperature given in fig. 26, the efficiency has been worked out for compression ratios between the limits given above when the maximum pressure is limited to 800 lb./sq. in., 1000 lb./sq. in., and also 1200 lb./sq. in., absolute. The results obtained are shown in fig. 29, which gives also the corresponding values for constant volume working. These curves show that the imposition of a fixed maximum pressure results in the

efficiency falling farther and farther below the constant volume figure as the compression ratio is increased, until it ultimately falls to the value for constant pressure conditions. The curve shows also that with any given maximum pressure there is a fairly definite limit to the ratio which may advantageously be employed, this limit being at about 15 : 1 when the maximum pressure is limited to 800 lb./sq. in., and 17 and 19 : 1 for maximum pressures of 1000 and 1200 lb./sq. in. respectively. Above these ratios the improvement in efficiency is so slight as not to justify any further increase, especially as under practical conditions the increase in average gas loading produced by the higher compression pressures will tend to increase the friction losses and so to nullify any improvement in efficiency which may be obtained on the indicated horse-power basis. Usually it will be found that there is not much to be gained in the brake thermal efficiency by increasing the compression ratio beyond about 14 : 1, but frequently there is a material improvement in smoothness and also in control over the maximum pressure, because, the delay period being reduced, there is a greater measure of control over the amount of constant volume burning.

It is not easy to determine practically the limit to which the compression ratio may be raised with advantage, because in changing the ratio it is impossible to avoid changing some other factor also. Changing the ratio results in changes in the form of the chamber and therefore influences the air movement. The spray characteristics, such as the size, dispersal, and penetration of the spray, may be influenced also, and it is thus impossible to differentiate between the effects of these several changes. The success of any particular design depends upon the balance obtained amongst the numerous factors, and a change in one of them may produce a change in another, and thereby completely mask the influence of the original alteration. This is particularly marked in engines with the smaller sizes of cylinder.

The analysis from which the results shown in fig. 29 were obtained was based upon the assumption that no heat is lost during combustion. The effect of loss of heat will be to increase slightly the maximum compression ratio possible for full constant volume combustion, and so give a small increase in the cycle efficiency. Above the critical ratio the maximum pressure will not be changed, because combustion conditions can be adjusted to maintain the chosen pressure and the heat loss therefore results in a reduction in the duration of the constant pressure part of the cycle, thus increasing slightly the mean expansion ratio, with a slight gain in cycle efficiency to offset a small portion of the cooling loss.

The effects of a change in ratio when there is a limitation to the maximum pressure are perhaps more readily understood if considered graphically, as shown in fig. 30. This represents, to scale, the pressure

volume changes considered in the above discussion when the maximum pressure is limited to 1000 lb./sq. in. From a practical standpoint we may conveniently examine the cycle from the point of view of the additional work which will be obtained by increasing the compression ratio to a figure above that which will allow the whole of the combustion to take place at constant volume, but at the same time leaving the maximum pressure unaltered. In fig. 30 *abcd* represents the diagram resulting from the constant volume combustion of the whole of the

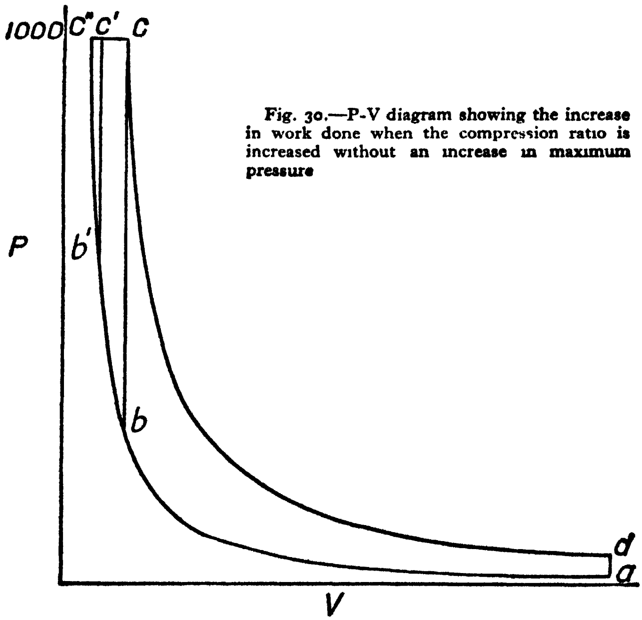


Fig. 30.—P-V diagram showing the increase in work done when the compression ratio is increased without an increase in maximum pressure

fuel, and the area of the diagram represents, to scale, the work which is done under these conditions, the compression ratio being 9 : 1. If the compression ratio is increased to 16 : 1 without the maximum pressure being increased, an additional quantity of work, equivalent to the area *bb′c′c*, will be obtained. This represents a quite appreciable increase. If, however, the compression ratio is raised yet further to the point where the compression pressure is equal to the maximum allowable pressure, the additional work developed will be represented by the area *b′c″c′*, a very small amount and one for the sake of which it will scarcely be worth while to incur any additional mechanical losses or difficulties of construction.

In fig. 31 are shown some actual figures connecting the maximum pressure and efficiency. The variations in pressure were obtained by changing the injection timing, and do not therefore represent the

full story, which can be obtained only by a change in the rate of injection. Advancing the injection beyond a certain point increases the loss caused by combustion commencing before top dead centre is reached and so masks some of the benefit from earlier combustion.

From the foregoing it will be clear that the necessity for limiting the maximum pressure places a definite limit upon the maximum efficiency attainable. Taking into consideration the conditions normally prevailing during compression, and making allowance for the change in specific heats, the maximum efficiency will be in the neighbourhood of 54 per cent if we accept 1000 lb./sq. in. as the maximum

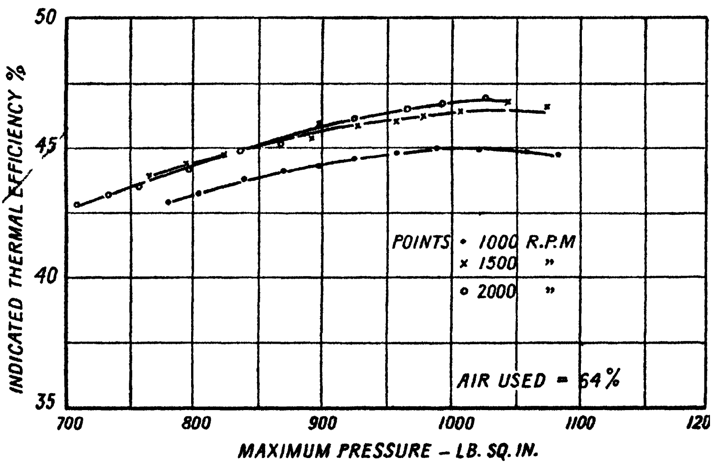


Fig. 31.—Influence of maximum pressure on efficiency

allowable pressure. The efficiency attainable with the same compression ratio if constant volume combustion is employed will be 56·8 per cent (fig. 29), i.e. a loss of 5 per cent is incurred. With a maximum pressure of 800 lb./sq. in., the maximum efficiency falls to 51·5 per cent, representing a loss of 9 per cent if the same figure 56·8 is taken for the constant volume figure. If the pressure is increased to 1200 lb./sq. in., the maximum efficiency will be increased to 56 per cent, an improvement of around 4 per cent, but at the same compression ratio the constant volume efficiency would have been 58 per cent, so that a loss of 3½ per cent is sustained. The increase in pressure to 1200 lb./sq. in. is obtained at an increase of 20 per cent in the maximum loading to be carried by every important part of the engine structure. Except in the case of quite small engines and those intended for special service where expensive materials can be employed, it is very doubtful whether the increase in loading would be justified.

6. The Effects of Heat Lost to the Cooling Water.

The whole of the heat carried away in the cooling water is frequently looked upon as the loss occasioned by the necessity for keeping the combustion chamber and cylinder walls within workable limits of temperature. This, however, is very far from being the actual case; of the total quantity of heat carried away in the cooling water only a comparatively small proportion is heat which could under any circumstances have been converted into useful work.

The air standard gives a figure for the fraction of the total heat supplied that the engine is capable of converting into work if no loss of any kind is sustained, and in its amended form we have the figure which would be obtained if allowance were made for the specific heats of the working medium. How the remaining heat is disposed of is quite immaterial from the point of view of the efficiency of the engine. It is lost in any case, and how it is lost does not make any difference to the engine. In the ideal engine the whole of the heat not converted into work is assumed to be rejected at the end of the cycle, i.e. in the exhaust gases in the practical form of the engine; but if some of the discarded heat is lost by other means it will make no difference to the efficiency of the engine. This actually occurs in the practical engine: a large amount of heat is lost from the cylinder and combustion chamber walls during the exhaust stroke, and some also from the exhaust port walls after the gases have actually left the working cylinder. All this heat appears in the jacket loss, but has no influence whatever upon the quantity of heat converted into useful work. Even the heat which is lost during the working stroke does not wholly represent an additional loss. The extent of the true loss depends upon where in the cycle the loss takes place. Heat lost during combustion will naturally have the maximum effect, while heat lost just before the end of the expansion stroke can have but very little effect, because its chances of doing useful work were very small. Even the heat lost during combustion does not represent a complete loss, because, even under the ideal conditions assumed for the air standard, only a part of this heat could be converted into work, and the balance would be rejected at the end of the cycle. Thus if the air standard efficiency is, say, 60 per cent, of every 10 units of heat lost during combustion, 6 only represents the true cooling loss, because the remaining 4 would have been rejected at the end of the cycle in any case.

It is a matter of some difficulty correctly to assess the heat loss at any particular point in the cycle, and any allocation is not much more than an estimate. Under full-load conditions the total quantity of heat carried away in the cooling water amounts to between 25 and 35 per cent of the heat supplied, the exact proportion varying with the type of engine and the speed. If we assume that half the total

quantity is lost during combustion and expansion it will be a fair assumption, because the surface and the time are in favour of the exhaust stroke although the temperatures are higher during the working stroke. This gives us, say, 13 to 17 per cent lost during combustion and expansion. Of this quantity, however, much is lost too late in the cycle to have done any useful work. Ricardo states that about half of this quantity is lost during combustion, and the remainder during expansion, and of this latter quantity he takes one-half as representing that which could have exerted a useful influence upon the work done. Using these figures, we have, say, 6 to 9 per cent lost during combustion, and the equivalent of 3 to, say, 4 per cent during expansion. Thus the total quantity of heat which reaches the water but which might otherwise have been usefully employed amounts to from about 9 per cent to 12 or 13 per cent of the total heat supplied, and of this heat only a part could have been converted into work. Assuming e.g. that the engine has an expansion ratio of 16, the efficiency on the air standard will be 67 per cent in round figures (fig. 24) and, allowing for change in specific heat, some 55 per cent (fig. 29) at full load; the quantity of heat which could have been converted to useful work, had all loss of heat to the cooling water been prevented, will therefore be from 6 to about 8 per cent under air standard conditions and from 5 to about 7 per cent when the properties of the working fluid are taken into account. That is, not more than 20 to 25 per cent of the heat carried away by the cooling water represents a true cooling loss, the remainder representing heat which would have been rejected in any case as part of the normal loss of the cycle itself.

The effect of a loss of heat during combustion is to reduce the maximum temperature, and therefore the effect of the change in specific heat also. Under constant pressure conditions the heat loss will mean that combustion will be completed somewhat earlier in the cycle and the adiabatic expansion will start sooner, with the result that under both constant volume and constant pressure conditions the heat lost during combustion results in a slight improvement in the cycle efficiency to offset some, if only a small part, of the loss.

7. The Effect of the Time required for the completion of Combustion.

The time taken before the combustion has been completed produces an effect akin to, but in reality quite different from, that of dissociation. Even gaseous mixtures, with an intimacy of contact between the fuel and the oxygen far greater than can be attained by even the finest of fuel sprays, have a rate of burning of a comparatively low order unless it is speeded up by some outside influence. The time required for combustion is such that under all circumstances some change in volume takes place while it is in progress. This is, of course, partly what we set out to do when operating on the constant pressure

cycle, but it is directly opposite to what is required for the constant volume cycle. As an appreciable measure of constant volume burning is an essential part of the cycle of operations of all high-speed compression-ignition engines, let us consider the effect of slowness of combustion upon constant volume burning.

Its effect is that the maximum pressure, instead of being produced when the volume is a minimum, will not be reached until after the piston has moved forward some distance on its working stroke. The pressure is therefore rising during the first part of the working stroke, from P_2, the compression pressure at V_2, the compression volume, to P_3' at volume V_2' a little way down the stroke, as shown in fig. 32. Had the combustion been instantaneous the pressure would have reached its maximum value P_3 at volume V_2, as shown by a dotted line in the figure, and an additional amount of work, equivalent to the area enclosed by the three points P_2, P_3, P_3', would have been done. The loss of this work involves a corresponding reduction in efficiency. In one respect, however, the reduction in the maximum pressure is an advantage if constant volume burning alone is employed, because it decreases the maximum loading which has to be taken into consideration in the design and so enables weight, and therefore cost, to be reduced. Where a certain amount of constant pressure burning is present, the time required for combustion causes a loss similar to that produced with full constant volume burning. The maximum pressure is not, or need not, be affected, because the rate of fuel delivery can be readjusted to produce the desired maximum, but the maximum pressure will not be reached until some time after top dead centre and constant pressure burning will continue further down the stroke, with a corresponding reduction in efficiency.

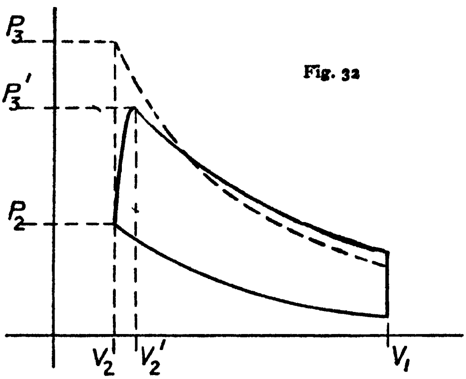


Fig. 32

In order that the maximum pressure shall not be reached too late in the stroke, the injection of the fuel is timed to begin shortly before the end of the compression stroke, and the rise in pressure due to combustion usually begins before top dead centre is reached. This produces a small increase in the negative work done during the compression stroke, and therefore represents a loss, but at the same time

the loss occasioned by the delay in reaching maximum pressure is reduced considerably and combustion is completed earlier, so that the net result is a quite considerable gain.

Normally the maximum pressure is reached between 10° and 15° of crankshaft rotation after top dead centre (T.D.C.). Less than 10° tends to produce rough running due to too high a rate of pressure rise, and at the same time may give too high a maximum pressure. More than 15° tends to increase the loss from the late burning. The actual forward movement of the piston for angular movements of the order just quoted is small, but with the high ratio of compression employed for high-speed compression-ignition engines, and the small compression volumes resulting therefrom, a quite material change in volume has occurred. With a normal ratio of connecting-rod length to crank throw the piston will have moved outwards about 1 per cent of its stroke when the crankshaft is 10° from T.D.C., and at 15° will have travelled slightly more than twice this amount; at greater angles the distance increases rapidly.

If the compression volume V_2 is given by the expression

$$V_2 = \frac{V_1}{(r-1)},$$

where V_1 is the initial volume and r the compression ratio, and if x is the increase in volume which occurs during the time required to reach the maximum pressure, then the volume V_2' at the point at which maximum pressure is reached is given by

$$V_2' = V_2 + x = \frac{V_1}{(r-1)} + x.$$

Hence the expansion ratio which takes place while the maximum pressure is being produced will be

$$r_p = \frac{V_2'}{V_2} = \frac{\dfrac{V_1}{(r-1)} + x}{\dfrac{V_1}{(r-1)}} = 1 + \frac{(r-1)x}{V_1}.$$

The values of r_p for a piston movement of from 1 to 5 per cent of its stroke are given in Table VIII for values of the compression ratio from 10 to 16.

That the influence of the increase in volume may be quite considerable is clearly indicated by the above figures, which also give an idea of the effect produced on the maximum pressure. The losses which may be occasioned if the maximum pressure is not reached quite shortly after T.D.C. are indicated also.

It must be pointed out, however, that these figures do not give

quite the whole picture, because the point of maximum pressure is not the point at which combustion is complete. The point of maximum pressure is nothing more than the point at which the fall in pressure occasioned by the forward movement of the piston (assisted by the heat lost to the cooling water) balances the increase in pressure produced by the heat of combustion. The point of maximum pressure does not necessarily coincide with the point of maximum temperature, and it will be found that at the higher load factors only a part of the combustion has been completed at the moment when the maximum pressure is reached. It is a matter of some difficulty to determine how far combustion has proceeded at any particular point in the

TABLE VIII

Expansion Ratio during Time required to reach Maximum Pressure and Expansion Ratio from Point of Maximum Pressure

Increase in Vol. (per cent) at point of Max. Press.	1	2	3	4	5	1	2	3	4	5
Approx. Crank Rotation deg.	10	15	18	21	23	10	15	18	21	23
Compression Ratio	Exp. Ratio to point of Max. Press.					Exp. Ratio from point of Max. Press.				
10	1·09	1·18	1·27	1·36	1·45	9·2	8·5	7·88	7·35	6·9
12	1·11	1·22	1·33	1·44	1·55	10·8	9·84	9·03	8·33	7·75
14	1·13	1·26	1·39	1·52	1·65	12·4	11·1	10·08	9·21	8·48
16	1·15	1·30	1·45	1·60	1·75	13·9	12·3	11·02	10·0	9·15

cycle, but a fairly general idea may be obtained by the study of suitable indicator diagrams, and it would appear that in burning around 80 per cent of the air, combustion is not completed until the crank has reached a point about 50 deg. after top dead centre, and the piston has travelled rather more than 20 per cent of its working stroke. The change in volume occasioned by a piston movement of this extent is far in excess of that required for maintaining constant pressure conditions. The increase in volume necessary for maintaining constant pressure conditions when the maximum pressure is restricted to 1000 lb./sq. in. lies between 1·3 : 1 to 1·6 : 1, according to the expansion ratio, while a change such as given above will mean an increase of around 3 : 1 or more, and the loss of effective expansion ratio is therefore considerable.

The total losses from slow burning probably amount to as much as 6 to 9 per cent.

8. Exhaust Losses.

In order that the exhaust gases may be rapidly and completely evacuated the exhaust valve is usually opened some little distance before the end of the expansion stroke. This causes the pressure to fall below the true expansion curve at the end of the stroke, but even with the early opening of the exhaust valve some pressure remains for a short time during the exhaust stroke, the two together involving a loss of work amounting to from 2 to 3 per cent.

9. The Effects of the Change in Volume produced by Combustion.

The change in volume caused by the rearrangement of the molecules during combustion is a factor which must be taken into account when assessing the probable efficiency of any given cycle. No such changes are considered under the idealized conditions, because no account is taken of the means whereby the heat is added to the gases. The change in volume may be positive or negative according to the fuel used; but in the case of the compression-ignition engine the change is positive. This means that at any point in the cycle after combustion has taken place the absolute pressure of the gases will be increased in proportion to the increase in volume. The work done during the expansion stroke is therefore increased in the same ratio and the efficiency of the cycle improved accordingly. It was shown in the account of the chemistry of combustion that the increase in volume for the chemically correct fuel-air ratio is nearly 7 per cent (p. 34). When 80 per cent of the oxygen is burned the increase is 4·8 per cent, and varies almost linearly with the quantity of air used. Strictly speaking, therefore, the cycle efficiency should be increased in the same proportion when making allowance for the various sources of loss.

10. Summation of Losses.

Summing up the losses given in the preceding paragraphs we have the following:

	per cent
Direct heat loss	9–13
Loss due to constant pressure burning	5– 9
Slow burning	6– 9
Exhaust loss	2– 3
	22–34

Against these losses must be set the gain from the increase in volume after combustion, amounting to nearly 5 per cent at the maximum load at 80 per cent air burned and falling to 2 per cent at the smaller load factors.

All the losses enumerated above decrease as the load decreases; the higher values may be taken as those occurring when 80 per cent

of the air is being used, and the net loss under these conditions may therefore be taken as around 28 per cent or, allowing for the variations of individual types, say 25 to 30 per cent. At the other end of the scale some diminution of the minimum losses may be expected under light load conditions, and the gross figure of 22 per cent may be reduced somewhat to perhaps 20 per cent or even to a trifle less, and with the gain from the increase in volume the net figure may be taken as varying from about 17 to 20 per cent. These figures do not, of course, take the mechanical losses of the engine into account.

CHAPTER V

The Air Charge before the Admission of the Fuel

1. Volumetric Efficiency.

The power of all internal-combustion engines is governed directly by the weight of air which can be passed through, and burned effectively in, the cylinder in unit time. A high volumetric efficiency is therefore a matter of supreme importance if a high specific output is to be obtained. This remark applies with added force to the compression-ignition engine, which at best is capable of utilizing only a part of the air it handles.

The volumetric efficiency of an engine is the ratio of the volume of the air it actually receives to the volume swept by the pistons. It may be expressed in one of two ways:

1. In terms of the air as measured at normal temperature and pressure, i.e. at 0° C. and 760 mm. mercury, in which case the result is known as the absolute volumetric efficiency;
2. In terms of the air as measured at the atmospheric conditions prevailing at the time of the test.

Clearly, the values given by these two methods may differ widely. Each, however, has its own particular sphere of usefulness.

2. Absolute Volumetric Efficiency.

To arrive at the absolute volumetric efficiency the quantity of air actually received is reduced to terms of N.T.P. As the existence of the standard conditions, especially in regard to temperature, is the exception rather than the rule, the value given by this method is usually lower than that given by the alternative method. The absolute value is used when making fundamental calculations, but for more general and practical purposes the alternative method is to be preferred.

3. Volumetric Efficiency in Terms of Prevailing Conditions.

This is arrived at by comparing the volume displaced by the piston and the volume of air received when measured at the temperature and

pressure of the atmosphere prevailing at the time of test. This gives a true measure of the breathing capacity of the engine, and is the only basis upon which different designs can be compared or the effect of a change in design determined. The engine can deal only with the air as it finds it, and to compare the absolute volumetric efficiencies of engines tested under perhaps widely differing atmospheric conditions is to do so on an entirely false basis, from which only misleading results can be obtained.

In addition to the effects of the prevailing atmospheric conditions, the weight of air received by the engine will be influenced by the following factors:

1. The resistance offered to the air during its passage into the cylinder.
2. The valve timing of the engine.
3. The amount of heat picked up by the air during its passage through the induction system.

4. The Resistance of the Induction System.

The induction system of a compression-ignition engine has one great advantage over that of a petrol engine; it has to handle air and air only. It can therefore be designed without the question of fuel distribution and deposition having to be considered. It has no carburettor to introduce resistance, and there is no need for any heating to assist in vaporizing the fuel. The system can therefore be made as generous as space will permit and the velocity of the air can be kept at a low figure.

A low velocity has two advantages: the frictional losses in the system are reduced, and at the same time the pressure drop required to produce the velocity is minimised. To keep the losses to a minimum the induction passages should be made as smooth as possible, and all bosses and projections avoided as far as is practical. The boss carrying the valve guide and the valve guide itself can add very materially to the resistance of the system and should be kept small, or avoided entirely by cutting back the guide until it is flush with the surface of the port and making good any deficiency in guide length by adding to the outer end of the guide. Sudden changes in the section, both in size and shape, should be avoided.

A condition peculiar to the compression-ignition engine is the necessity, with certain types of combustion chamber, for providing a mask or baffle around part of the perimeter of the valve in order to produce a rotational or swirling movement of the air within the cylinder. It might be thought that this mask would have a very detrimental effect upon the breathing of the engine, but in actual practice it can be made to offer comparatively little resistance, and therefore has only

quite a small influence upon the volumetric efficiency. This is in part due to the comparatively restricted rotational speed of most compression-ignition engines; at high speeds the presence of the mask would certainly produce a more marked effect.

Compression-ignition engines which are called upon to operate at the higher speeds are chiefly those intended for vehicle propulsion. As these are mostly the larger sizes of automobile engine, they are usually limited to a maximum speed of not more than 2000 r.p.m. or a trifle over, and in many cases have a maximum speed of several hundred r.p.m. less. At speeds of this order the maximum velocity through the valves is usually moderate, and the presence of the mask therefore does not increase the velocity to a value likely to produce a very large pressure drop.

In addition to this, the position normally occupied by the mask is a factor helping to reduce the resistance it offers. The cylinder diameters normally associated with the higher speeds are somewhat on the small side, usually under 5 in. diameter, and in the endeavour to provide all the breathing capacity possible the valve diameters are increased to the maximum allowed by the various features of the design. This brings the valve in close proximity to the cylinder wall at one side, the clearance between the valve head and the cylinder wall frequently being reduced to the minimum safe figure. This close proximity of the valve and cylinder wall has a pronounced influence upon the air flow, even though no mask is provided. The cylinder wall itself masks a quite appreciable part of the circumference of the valve, as will be seen from fig. 33, which is drawn more or less to scale and shows that nearly one-fourth of the circumference is inoperative owing to the proximity of the cylinder wall. Contrary to what one would expect, however, the effect of this blanking of the valve is actually *beneficial*, and the total pressure head required to produce a given rate of air flow is less than that required when the valve is placed centrally in the cylinder. Experiments for measuring the air flow through an engine from which the piston has been removed show that for a given flow of air the total pressure head decreases as the valve is moved away from the centre of the cylinder and reaches a minimum when the valve is right against the cylinder wall.

The reason for this apparent anomaly is not difficult to determine. The air discharged from the valve takes the form of a flat cone, and when the valve is placed centrally in the cylinder the air stream strikes the cylinder wall in a direction very nearly normal to the surface, as shown in fig. 34. Serious eddying is thus produced, and this increases the resistance to flow or, what is really the same thing, precludes any recovery of the velocity head into pressure. If, however, the valve is placed against the cylinder wall, although a fair proportion of the circumference may be masked, the flow from the remainder strikes

the cylinder wall at a much more favourable angle (fig. 35), and is deflected downwards into the cylinder with much less eddying and a fair chance of some pressure recovery.

The experiment of introducing a little dye or ink into the air stream above the valve when the valve is in the two positions illustrates the point very clearly and indicates the orderly flow of the air down the cylinder when the valve is placed well to one side of the centre.

Under such circumstances it is not difficult to imagine that the presence of an additional mask attached to the valve itself does not necessarily mean a large increase in the resistance of the system. The function of the mask is to direct the flow to one side of the centre of

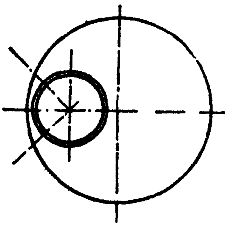


Fig. 33

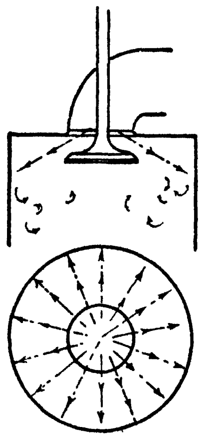


Fig. 34

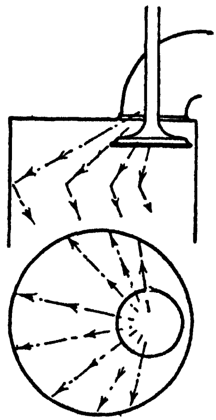


Fig. 35

the cylinder and so produce a rotary movement of the air. This does not necessarily mean that the whole of the air must be made to pass to one side of the centre of the cylinder; any lack of uniformity of flow about the centre will produce a rotary movement, although under some circumstances it may be necessary to direct the whole of the air stream to one side of the centre if the desired result is to be attained. A comparatively small mask will sometimes be found to provide all the rotary movement required. A mask occupying only 45° of the circumference of the valve has been found sufficient in certain cases, while an engine with a mask occupying 90° is in large scale production.

The presence of the mask tends to assist in producing an orderly flow of the air into the cylinder by removing, in part at least, the opposing stream from the opposite side of the cylinder, and so reducing the eddies produced when the two streams meet; the air then flows in with a sweeping movement free from much of the indiscriminate turbulence which is normally produced.

A mask extending for as much as 180° around the valve does not produce anything approaching the restriction that might be supposed. Fig. 36 is an illustration of the effects produced by masks of different

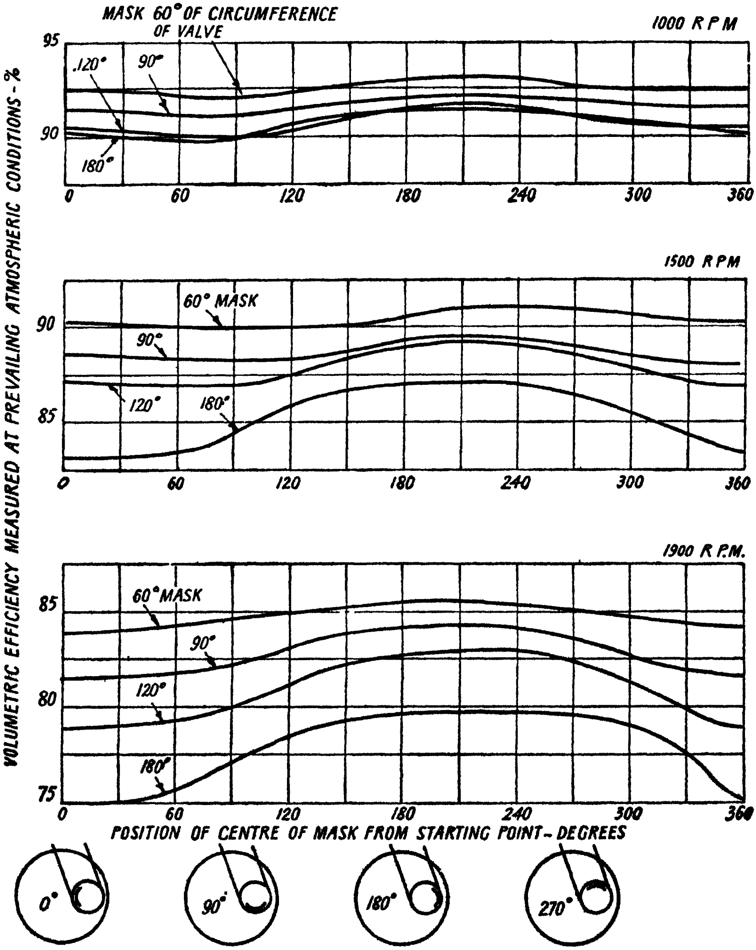


Fig. 36.—The influence of the size and position of the mask upon the volumetric efficiency of the engine

sizes at a number of different engine speeds, the curves showing the volumetric efficiency of the engine when the mask was in different positions obtained by rotating the valve.

5. Valve Diameter and Lift.

The question of valve diameter does not call for much discussion, as in the case of high-speed engines it is pretty well standard practice to provide valves as large as the diameter of the cylinder will permit. The presence of the injector nozzle is sometimes a disturbing influence, but much can be done by placing the valves and the nozzle slightly off centre in opposite directions. An expedient sometimes adopted is to machine recesses in the cylinder walls to allow valves of a larger diameter to be fitted. This, however, has the disadvantage of separating part of the clearance volume from the main combustion chamber, with the result that the air contained therein is not in a position to be easily reached by the fuel. On this score the arrangement is to be avoided if at all possible, although it must be admitted that the method has been adopted by one of the most successful British manufacturers. There is an added difficulty when thin "dry" cylinder liners are used.

The comparatively restricted speeds of the compression-ignition engine enable advantage to be taken of a high lift-diameter ratio. The resistance offered by the valve decreases up to a lift more than one-half the diameter, although there is not much to be gained by going beyond $\frac{5}{8}$ diameter.* Where the diameter is at all restricted, as when a long stroke-bore ratio is employed, a lift up to $\frac{3}{8}$ diameter may be used with advantage without excessive rates of acceleration having to be resorted to. At the same time quick lift cams can be used with a smaller lift-diameter ratio without increasing the acceleration unduly. The author has found that an inlet air speed of 250 f.s., calculated upon the throat diameter of the valve at 2000 r.p.m., will give adequate breathing up to that speed when the lift of the valve is made one-third of the throat diameter.

The necessity for taking every reasonable step to improve the breathing will be more fully realized when it is pointed out that the quantity of fuel received by the engine is not, as is the case with the petrol engine, governed by the quantity of air it receives. In the petrol engine the quantity of fuel received per stroke falls off with the quantity of air, as the volumetric efficiency decreases at the higher speeds. Not so in the compression-ignition engine; the fuel is delivered by an independent metering device, the fuel pump, and it is a difficult matter to arrange the full-load delivery of the fuel pump to match a badly drooping volumetric efficiency curve. To avoid uneconomical operation and a dirty exhaust the delivery from the fuel pump must be adjusted to deliver no more fuel than the engine can economically consume at the maximum speed at which it is intended to run. If

* *National Advisory Committee for Aeronautics, U.S.A.*, Report No. 24.

the volumetric efficiency at the maximum speed has fallen much below that obtained at the lower part of the speed range, the maximum fuel delivery to the engine will be handicapped throughout the whole speed range, and the performance of the engine will suffer.

Provided that the position of the injector nozzle does not interfere, the exhaust valve may advantageously be reduced in diameter for the purpose of obtaining a larger inlet valve. The size of the exhaust valve is comparatively unimportant, because under full-load conditions the presence of a considerable pressure at the end of the working stroke ensures that the bulk of the exhaust gases is evacuated during the few degrees following the opening of the exhaust valve. A pressure of 45 lb./sq. in. when the exhaust valve opens means that 75 per cent of the weight of gases is discharged in the rush of gas as the pressure falls to that of the atmosphere. The small density of the remainder makes it offer very little resistance in being pushed out of the cylinder. The area of the exhaust valve may therefore be made 70 per cent, or even less, of the inlet valve area and the inlet valve increased in size if this can be done without interfering with the optimum position of the injector nozzle.

6. The Effects of Valve Timing.

The timing of the valves has a direct bearing upon the maximum volumetric efficiency. The point of opening of the inlet valve is not of very great moment, provided that it is not opened so early as to allow exhaust gases to pass out into the inlet manifold and so be returned again to the cylinder during the suction stroke.

It is the closing point of the inlet valve which is the important factor and one which can exert a marked influence upon the volumetric efficiency. In all high-speed engines it is usual to close the inlet valve at an appreciable angle after bottom dead centre has been passed. This allows the valve to retain a good portion of its lift right up to the end of the piston stroke and thus allows air to flow in during the "dwell" of the piston at the bottom dead centre. If, however, the valve is not closed until the piston has travelled back some distance on the compression stroke, the maximum possible quantity of air which can be retained by the engine will be reduced by the volume swept by the piston between the bottom dead centre and the closing point of the valve. The closing point normally adopted for high-speed engines is between about 35° and 40° after bottom dead centre, and with angles of this order the piston has travelled up on the compression stroke a distance sufficient to make a quite appreciable difference to the volume of air retained in the cylinder. The extent of the movement of the piston will depend upon the connecting rod/crank-throw ratio, but does not vary much over the ratios normally adopted

for high-speed engines, as will be seen from the figures given in Table IX.

TABLE IX

Inlet Valve closes deg. after B D.C.	0	10	20	30	35	40	45	50
Ratio Rod/Crank	Volume retained in Cylinder							
3·5	100	99·5	97·8	95·1	93·3	91·2	88·9	86·3
4·0	100	99·5	97·7	94·8	93·0	90·9	88·5	85·8
4·5	100	99·4	97·6	94·7	92·8	90·6	88·1	85·4

From these figures it will be seen that with the point of valve closing normally used for high-speed engines the maximum possible volumetric efficiency, measured at the prevailing atmospheric conditions, is from 90 to 93 per cent. Any volume of air in excess of this figure which may have been drawn into the cylinder during the suction stroke will be rejected through the inlet valve during the first part of the compression stroke.

The closing point of the inlet valve must therefore be chosen to give the best all-round results for the service for which the engine is intended. For an engine intended to run at one set speed or, as in the case of a marine or aircraft engine, to develop its maximum output at one predetermined speed only, it is merely a matter of determining either experimentally or from past experience the closing point that will give the maximum degree of filling at that one speed.

For engines which are required to operate over a wide range of loads and speeds, a compromise must be made between the high and low speed ends of the speed range. A late closing made in the interests of the higher speeds will result in some loss in torque at the low speed end, while an early closing in the interests of the low-speed torque will result in a more rapid falling-away in power at the higher speeds.

The closing point of the exhaust valve is a matter of some importance. Too early closing may result in some of the exhaust gases being trapped. When the inlet valve opens this gas will escape into the inlet manifold to be drawn back again into the cylinder. This may be avoided by closing the exhaust valve a few degrees after top dead centre is reached. This, it might be argued, tends in the same direction, but with the figures normally employed, up to about 10 deg., not more than 1 per cent of the swept volume can be lost. This loss is small compared with the possible effects of closing the valve too early, and an additional safeguard is derived from the fact that the direction of flow of the exhaust gases has first to be reversed before any can be drawn back into the cylinder.

From the foregoing it will appear that under average conditions the volumetric efficiency, measured at the prevailing conditions, has a maximum possible value of about 92 per cent.

7. The Effects of Heat picked up during Induction.

The amount of heat picked up by the air during admission influences the weight of air obtained by the engine in exactly the same way as an increase in atmospheric temperature does. The extent of this heating is less in a compression-ignition engine than in the case of a petrol engine, owing to the much lower temperature normally reached by the various parts of the cylinder; further, we are not, as in the petrol engine, under the necessity of adding heat for the purposes of assisting vaporization. On the other hand, there is the loss of any

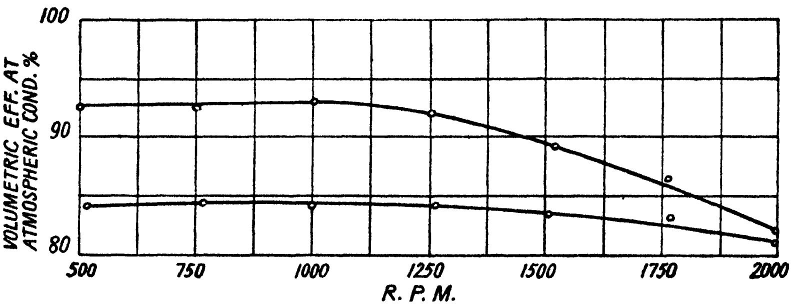


Fig. 37.—Variation in volumetric efficiency with speed at full load and also when motored cold

advantage which the petrol engine may derive from the refrigeration effects produced by the latent heat of evaporation of the fuel, but at the same time there is no displacement of air by the fuel vapour so formed. It is only the heat picked up from contact with, or radiation from, the hot metal surfaces which has any influence upon the volumetric efficiency; the heat received from the mixing of the fresh air with the residual exhaust gases has no effect upon the volumetric efficiency, because the exhaust gases lose in cooling a volume exactly equal to that gained by the fresh air from being heated by exhaust gases.

The amount of heat picked up by the air depends on both the load and the speed at which the engine is running. The temperature of the cooling water also exerts a measurable effect. As the engine speed increases the time that the air is in contact with the hot surfaces decreases, so that the amount of heat it receives is reduced. On the other hand, for any given brake mean effective pressure the temperature of the cylinder walls, &c., will increase as the speed increases, and there is thus a tendency for a larger amount of heat to be transmitted to the air. These two effects do not, however, quite cancel out, and the net

result is for the heating of the air to decrease as the speed increases. The difference is sufficient to have a marked effect upon the breathing capacity of the engine, and tends to offset the increasing resistance offered by the induction system as the air velocity increases. The decrease in volumetric efficiency as the speed increases is thus smaller than would otherwise be the case, and a much flatter speed-volumetric

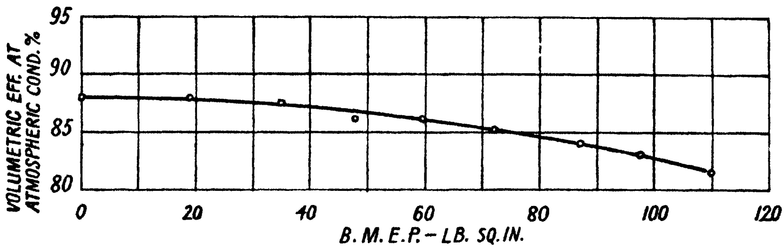


Fig. 38.—Variation in volumetric efficiency with load at 1500 r.p.m.

efficiency curve is obtained when under power than is obtained when the engine is motored in a cold state. This is shown in fig. 37, which gives a typical pair of curves.

The effect of a change in load at a fixed speed is to decrease the quantity of heat picked up as the load decreases. This is what we should anticipate, but the difference is rather more marked than might perhaps be expected. A typical example of the effects of a change in

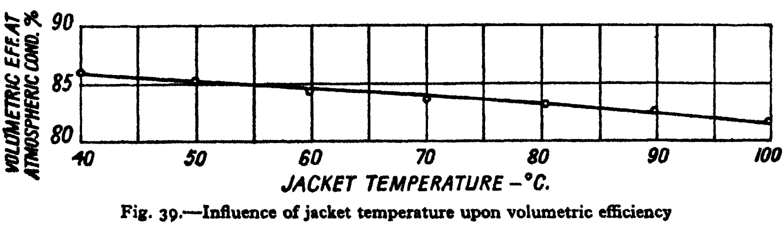


Fig. 39.—Influence of jacket temperature upon volumetric efficiency

load is shown in fig. 38, while that produced by a change in cooling water temperature at a fixed load and speed is shown in fig. 39.

The extent of the heating will vary with the design of the engine; engines with a long induction system naturally tend to heat the air more than those having a shorter system. Engines having a high cylinder wall temperature will have the same tendency. Speaking generally, an increase in air temperature during induction of between 20° and 30° C. appears to be the normal range for high-speed engines when running under full load, the higher figure applying to speeds of from about 800 to 1400 r.p.m., the lower figure to speeds approaching 2000 r.p.m.

Direct measurement of the rise in air temperature is not possible, but it may be obtained indirectly and with sufficient accuracy for all practical purposes if measurements are made of the volumetric efficiency of the engine under normal operating conditions and again while the engine is being motored at the same speed and in a cold state. The ratio of the absolute temperatures of the air before and after admission will be the inverse of the ratio of the volumetric efficiencies measured under the two conditions. Thus, if an engine when receiving air at 17° C. (290° abs.) has a volumetric efficiency of 90 per cent when motored cold at a certain speed, and of 82 per cent running at full load at the same speed, then the temperature of the air after admission, but before mixing with the residual exhaust gases, will be $290 \times 90/82 = 318°$ abs., an increase of 28° C. during admission.

It should be pointed out that there is a possible source of error in this method if the valve timing is such as to allow exhaust gases to be trapped at the end of the exhaust stroke. The volume of gases so trapped will vary with the load and speed and under certain conditions may be greatest when the engine is motored. With suitable valve timing, however, this should not occur. Another possible source of error occurs if the exhaust system is such that a scavenging effect is produced when running under load. This, however, is not likely with multi-cylinder engines fitted with a common exhaust manifold.

In section 6 (p. 86) it was stated that with the valve timing normally employed for high-speed engines the maximum possible volumetric efficiency measured at the prevailing atmospheric conditions is around 92 per cent. In order to arrive at the absolute volumetric efficiency this figure must now be corrected for the heating of the air during induction. Assuming, for the moment, that the atmospheric conditions are standard, i.e. 0° C. and 760 mm. mercury, the maximum possible absolute volumetric efficiency will vary from $92 \times 273/293 = 85{\cdot}7$ per cent to $92 \times 273/303 = 83$ per cent when the temperature rise during induction varies from 20° C. to 30° C., as already given. At the more normal air temperature of 17° C. these figures will be further reduced to between 81 per cent and 78·5 per cent, and with still higher atmospheric temperatures the figure will be reduced yet more.

The influence of the barometer is in direct proportion to the barometric height, and will be plus or minus according as the barometer stands high or low. Since, however, the barometer normally is not very far removed from the standard figure, we may say that the maximum absolute volumetric efficiency is about 80 per cent. The above remark about the barometer must not be construed to mean that the influence of the barometer is unimportant, as this is certainly not the case, especially when an engine is working towards the limit of its output at any given speed. Under these conditions a change in

the barometric height may make a quite appreciable difference. When, however, the engine is working well within its limit the small change in excess air produced by a change in barometric height will make little or no difference to its performance.

With many installations a certain amount of artificial heating of the air takes place before it reaches the engine intake, and it usually happens that the air received by the engine is at a rather higher temperature than the atmosphere. The extent of this heating will vary greatly with the nature of the installation, and its importance will depend largely upon how near to the limit of a clear exhaust the engine is called upon to operate. A stationary engine in a well-ventilated engine-house will be subject to comparatively little variation in temperature, which will remain much the same summer and winter. An engine under the bonnet of a commercial vehicle or a tractor will have very wide variations, which may run from —10° C. and even less in winter, to 50° C., and even more, in summer, although under average running conditions the difference will be much less than this. The latter temperature is due to the engine receiving air which has first passed through the radiator and has therefore received a great deal of heat, and under such circumstances it may be desirable to arrange for the engine to receive air which has not been preheated to so great an extent. The air will undergo a certain amount of heating in the neighbourhood of almost any engine, and under normal conditions it is probably safe to say that the air received by the engine has a temperature of from 20° to 25° C. Taking the temperature increase during induction as being on the average 25° C., we arrive at a figure of from 318 to 323° abs., say 320° abs., as the average temperature of the air after admission, but before mixing with the exhaust gases. At this temperature the absolute volumetric efficiency will be $92 \times 273/320 =$ 78·5 per cent, which may be taken as a maximum, at standard barometric height, for general operating conditions. Taking everything into consideration, 75 per cent is probably a safe figure upon which to base calculations.

8. The Temperature at the Commencement of Compression.

The temperature at the commencement of compression will be governed by the temperature of the air after admission, the temperature of the residual exhaust gases, and the relative weights of the two. It has just been shown that under average conditions the temperature of the air may be taken as being 320° abs. The temperature of the exhaust gases will vary greatly with the load upon the engine and the expansion ratio. Under conditions of maximum output, i.e. when the maximum possible quantity of air is used, temperatures of between 400° and 500° C. are reached, although under conditions calling for continuous operation the lower figure is nearer the normal

value. If, therefore, we take a figure of 700° abs. for the exhaust temperature at maximum load, we shall not be far out.

If we work on the assumptions made when discussing the volumetric efficiency, the charge at the end of the suction stroke will consist of 92 per cent of the swept volume of fresh air at 320° abs., and a volume of exhaust gases equal to the clearance volume and at a temperature of 700° abs. At a compression ratio of 16 : 1 the volume of the exhaust gases will be $100/(16-1) = 6{\cdot}67$ per cent of the total volume at a temperature of 700° abs.

The relative amounts of the two gases will then be $92 \times 273/320 = 78{\cdot}5$ parts by weight of air and $6{\cdot}67 \times 273/700 = 2{\cdot}6$ parts by weight of exhaust gases.

If we neglect the difference in the specific heats of the two gases (this will be comparatively small at the temperatures under consideration) the temperature of the mixture will be

$$\frac{78{\cdot}5 \times 320 + 2{\cdot}6 \times 700}{(78{\cdot}5 + 2{\cdot}6)} = \frac{25{,}120 + 1820}{81{\cdot}1} = 332^\circ \text{ abs.,}$$

say 330° abs.

At a lower compression ratio the volume of exhaust gases will be increased and the temperature also will be somewhat greater, but with any compression ratio likely to be used for a compression-ignition engine the total volume of exhaust gases will be small, their effect upon the final temperature will be small also, and the variation in final suction temperature with compression ratio not very great. Taking everything into consideration, the temperature at the beginning of compression will be found to be between about 330° abs. and 340° abs. for full-load conditions. Under no-load conditions, after making allowance for the improved volumetric efficiency obtained under these conditions and also the greatly reduced exhaust temperatures, the corresponding figures work out at between 300° and 310° abs.

9. The Effective Compression Ratio.

The compression ratio is commonly taken as being the ratio of the volume given by the sum of the swept volume and the clearance volume, to that of the clearance volume alone. In order to arrive at the resulting compression pressure the procedure usually adopted is to utilize the ratio thus determined and to assume that under running conditions the pressure within the cylinder when the piston is at the outer limit of its travel is from 13·0 to 13·5 lb./sq. in. abs. This is a somewhat empirical method, but it gives results which agree quite well with the pressures actually found in practice.

An alternative method, which has, perhaps, the merit of being somewhat less empirical, is to ascertain the compression ratio from the

position occupied by the piston at the moment the inlet valve closes, and to assume that the pressure within the cylinder at this point is equal to that of the atmosphere. This assumption is certainly justified if the valve timing has been correctly chosen for the operating speed of the engine, and experience indicates that, in the case of engines intended to operate at varying speeds, atmospheric pressure is reached at the closing point of the inlet valve at speeds in the neighbourhood of that at which the engine develops its maximum torque. At higher speeds, when the volumetric efficiency has fallen off somewhat, the pressure will be below that of the atmosphere at the closing point of the inlet valve and a suitable allowance must then be made.

As has already been pointed out, the average closing point of the inlet valve of a high-speed engine reduces the volume of the air retained in the cylinder to a maximum value of about 92 per cent of the volume swept by the piston. The alteration in compression ratio due to the ratio being measured from the closing point of the inlet valve is as shown in Table X.

Table X

Compression Ratio from B.D.C.	Swept Volume	Clearance Volume	Volume swept from inlet closing point	Compression Ratio from inlet closing point
10	100	11·1	92	9·3
12	100	9·1	92	11·1
14	100	7·69	92	13·0
16	100	6·66	92	14·8
18	100	5·88	92	16·6

A comparison of the compression pressures calculated in the more usual way and those calculated as suggested above is given in Table XI, the value of γ, or rather n, being taken as 1·35 for both.

Table XI

Compression Ratio		Compression Pressures lb./sq. in. abs.		
From B.D.C.	From Inlet Closing	From B.D.C. Ps. = 13	From B.D.C. Ps. = 13·5	Ratio measured from Inlet Closing
10	9·3	291	302	298
12	11·1	372	387	379
14	13·0	458	476	469
16	14·8	549	570	559
18	16·6	643	668	652

Ps. = assumed suction pressure at B.D.C.

The difference in the results obtained from the two methods is small, the pressure obtained by the use of the "effective compression

ratio " being about the same as the more usual method would give if the final suction pressure were taken as 13·25 lb./sq. in. abs.

At first sight it would appear that on account of the comparatively large difference in the value of the compression ratio a much more serious difference between the two methods would be produced in the case of the compression temperatures. The compression temperature, however, is governed by the compression ratio raised to the power of $\gamma - 1$, and as the value of the index is small the temperature does not increase very rapidly for small changes in r and the difference in the results from the two methods is not serious.

It is clear, however, that fundamentally it is more correct to use the effective compression ratio, but that the difference in result is not serious is shown by the figures in Table XII, which gives the compression temperatures calculated both ways for the same conditions as for Table XI, the final suction temperature being taken as 340° abs. The differences shown are no more than would be produced by a slight difference in the value chosen for γ, or in the assumptions made when calculating the final suction temperature. From this it might be argued that if the difference in the results is so small the more usual method is all that is needed, and there is no object in using the other. It must be admitted that the older method does serve a very useful purpose, in that it enables results to be worked out when, as is frequently the case, only the compression ratio based upon the full swept volume has been given. The use of the " effective compression ratio ", however, takes into account an important feature of the design, namely, the valve timing, and at the same time the calculations are based upon a more accurate basis; this method, therefore, is to be preferred for design and experimental work.

TABLE XII

r measured from B.D.C.	Compression Temp. deg. abs.	Effective Compression Ratio	Compression Temp. deg. abs.
10	760	9·3	751
12	802	11·1	790
14	855	13·0	834
16	895	14·8	874
18	935	16·6	909

10. The Value of γ or n during Compression.

For a truly adiabatic change in volume the value of the exponent n, in the equation $PV^n = \text{Constant}$, is equal to γ, the ratio of the two specific heats of the gas under consideration. All theoretical calculations connected with internal-combustion engines assume that the

working fluid is pure air, and the value of γ is taken as 1·4. Under actual operating conditions, however, truly adiabatic changes in volume are the exception rather than the rule, and factors other than the ratio of the two specific heats of the gas enter into the relationship between P, V and T; it is therefore usual to denote the exponent by n rather than γ when dealing with practical engine conditions. The exponent n may be considered as the apparent value attained by γ under working conditions, and by making allowance for the various influences, it enables results to be obtained by calculation which agree with those obtained practically.

The exact value of n will be governed by the mean specific heats of the gas over the working range of temperature and also by the heat gained or lost during the changes in volume. As is explained more fully elsewhere, the specific heats of gases do not have a constant value but change with temperature, and at the same time the rate of gain or loss of heat during a change in volume is not constant but varies from point to point during the cycle. The whole question is therefore very involved, and the problem is that of choosing a value for n which will most nearly give results agreeing with the actual operating conditions.

During the compression stroke the gases may be considered as composed of air only. The quantity of residual exhaust gases is very small in the case of the compression-ignition engine, and of this small quantity at least 20 per cent is air unused during the previous cycle. The change in specific heat may therefore be taken as being the same as for air.

The temperature range normally covered by the compression stroke under working conditions runs from not less than about 300° abs. at the commencement of the stroke to 1000° abs., or a trifle over, at the end of the stroke. For this range the mean ratio of the two specific heats may be taken as 1·39. The variation occurring over this range of temperature is not very great.

The gain or loss of heat during compression is a rather more difficult matter to determine. It is, however, fairly safe to say that generally speaking heat is received by the air during the early stages of compression, but is lost from it during the final stages. The relative magnitudes of the gain and the loss will vary with different engines, and will depend very largely upon individual conditions. These conditions include the nature and extent of the movement of the air during the compression stroke and the presence or otherwise of hot surfaces with which the air can make contact while undergoing compression. The influence of the latter is very marked and may produce wide variations in the final compression pressure resulting from a given compression ratio.

The speed of the engine has a very definite influence upon the gain

or loss of heat during compression. As the speed increases the time factor is reduced and less heat will be picked up during the early stages of compression, but less heat will be lost during the later stages also. The velocity of the gases over the cylinder walls will increase with the engine speed, and so tend to increase the rate of heat exchange in both directions. At the higher speeds the temperature reached by the cylinder walls at any given B.M.E.P. will be greater than for the same B.M.E.P. at a lower speed, and so tend to produce a higher temperature at the end of compression.

The whole situation is thus very complex, but, generally speaking, there is a tendency for the compression temperature to increase with the engine speed, and the increase in temperature is frequently sufficient to produce a marked increase in compression pressure, and, in some instances, even though there has been some falling-off in the volumetric efficiency. This condition is particularly noticeable in the case of combustion chambers which are provided with a heated surface from which the air may extract heat during the compression stroke, and indicates that there must be a marked increase in the value of n as the engine speed increases. Engines not specially provided with a hot surface may show this tendency also, but to a much smaller extent. This feature will be discussed in more detail later; strictly speaking, it is a feature which arises before the admission of the fuel, but as it has a very important bearing upon combustion conditions it may with advantage be discussed under that heading.

A coating of lime on the water side of the combustion chamber walls will result in a material increase in the wall temperature; this will affect the heat losses during compression and hence the value of n.

With such wide variations it is obviously a matter of some difficulty to determine the precise value for n; in fact, it will be clear that n can have no precise value but will depend upon individual circumstances. We must, therefore, consider the limits between which the value of n may reasonably be expected to lie.

For petrol engine calculations n is usually taken to have a value of from 1·30 to 1·35. In petrol engines, however, the presence of fuel in an unvaporized condition introduces another factor. A quite appreciable proportion of the fuel is vaporized during the compression stroke and the latent heat of evaporation absorbs part of the heat of compression and so reduces both the compression pressure and the compression temperature.

No such effect is present in the compression-ignition engine, so that in these engines n may reasonably be expected to have a value somewhat higher than that for the petrol engine, although the higher temperatures reached during the later stages of compression accentuate the heat loss as compared with engines having a lower compression ratio.

The author has measured compression pressures which indicate values of n as high as, and even slightly in excess of, 1·4. This, however, was with combustion chambers provided with a heated surface for the express purpose of increasing the compression temperature. With open combustion chambers at moderate speeds the value 1·35 appears to be fairly representative, but at the higher speeds values above this are frequently found, while at the lower speeds a somewhat smaller value may be found. Taking everything into consideration, 1·35 seems to be a fair average value for chambers not provided with special heating arrangements. In the latter, values somewhat higher than 1·35 may normally be expected, and about 1·375 would appear to be a good all-round value, although at high speeds and loads 1·4 may sometimes be reached.

The size of the engine cylinder is a factor which has some bearing on the results. With a small cylinder the heat losses are increased, because the surface-volume ratio is less favourable than in the case of a larger cylinder, and frequently it is found necessary to provide a small cylinder with a somewhat higher compression ratio in order to compensate for the heat loss.

Tables XIII and XIV give P_2/P_1, the ratio of pressures before and after compression, and T_2/T_1, the ratio of corresponding temperatures, for compression ratios from 9 to 20, and values of n from 1·30 to 1·40.

TABLE XIII

Ratio of Absolute Pressures before and after compression (P_2/P_1)

r	n				
	1·300	1·325	1·350	1·375	1·400
9	17·4	18·4	19·2	20·5	21·7
10	19·9	21·1	22·4	23·7	25·1
11	22·5	23·9	25·5	27·0	28·7
12	25·3	26·9	28·6	30·4	32·4
13	28·0	30·0	32·0	33·9	36·3
14	31·0	33·0	35·2	37·6	40·2
15	33·8	36·1	38·7	41·5	44·2
16	36·8	39·5	42·2	45·3	48·5
18	42·8	46·1	49·5	53·2	57·2
20	49·1	52·9	57·1	61·6	66·3

The above figures are to the nearest first place of decimals.

TABLE XIV

Ratio of Absolute Temperatures before and after Compression (T_2/T_1)

r	n				
	1·300	1·325	1·350	1·375	1·400
9	1·93	2·04	2·13	2·28	2·41
10	1·99	2·11	2·24	2·37	2·51
11	2·05	2·18	2·32	2·46	2·61
12	2·11	2·25	2·38	2·54	2·70
13	2·16	2·30	2·45	2·61	2·79
14	2·21	2·36	2·52	2·69	2·87
15	2·25	2·41	2·58	2·76	2·95
16	2·30	2·46	2·64	2·83	3·03
18	2·38	2·56	2·75	2·96	3·18
20	2·46	2·65	2·86	3·08	3·31

The above figures are to the nearest second place of decimals.

Since the engine depends upon the temperature of the air for the ignition of the fuel, the value of n is almost as important as the compression ratio, as an examination of Table XIV will show. To quote an example, if we assume that the initial temperature remains at a figure of 300° abs. at all values of compression ratio (an assumption not very far from the truth), then in order to obtain a final compression temperature of, say, 775° abs., the compression ratio required would vary from 16 : 1 for a value of $n = 1·30$ down to something less than 9 : 1 for a value of $n = 1·40$. As is explained later, however, the temperature alone is not the sole criterion, density also being an important factor, and the example just given does not give quite the true picture. It does, however, serve to show what a wide variation in compression temperature can be caused by changes in the value of n.

The actual conditions of pressure and temperature reached during the final stages of the compression stroke are of vital importance to the operation of the engine. They do, in fact, determine the satisfactory nature, or otherwise, of the engine's performance. As, however, they are so intimately connected with the sequence of events following the introduction of the fuel, it is felt that they may best be discussed when that phase of the cycle of operation is being considered.

CHAPTER VI

The Process of Combustion in the Compression-Ignition Engine

1. The Method of Adding Heat to the Charge of Air.

In all practical internal-combustion engines the addition of the heat must be progressive if dangerous shocks are to be avoided, and if the requisite degree of smoothness of operation is to be obtained. The instantaneous addition of heat assumed to take place in the idealized constant volume cycle, and also during the constant volume part of the mixed cycle, is neither possible nor desirable for the practical engine. What is required is a controlled rate of burning of the fuel that will give a high efficiency but at the same time will avoid both excessive maximum pressures and undesirable rates of pressure rise.

In practical engines two methods of introducing the fuel are available. It may be introduced with the charge of air or it may be introduced independently from the air at any desired point in the cycle.

2. The Simultaneous Introduction of Fuel and Air.

In engines to which the fuel and air are introduced simultaneously, the correct proportions of fuel and air are maintained automatically by means of some form of carburettor or similar device. The range of suitable fuel/air mixtures is a comparatively narrow one, and a fairly close control over the ratio is essential. The fuel and air are then compressed together, and are ignited by suitable means at the desired point towards the end of the compression stroke. Once ignition has taken place, burning proceeds rapidly and its rate is governed by the proportion of fuel to air, the properties of the fuel itself, the degree of turbulence, the temperature and pressure within the cylinder, and the shape and size of the combustion chamber. No external control, other than the point at which ignition is to take place, is possible, and we have therefore to depend upon the choice of fuel and the design of the combustion chamber.

The mixture being compressed adiabatically, its temperature increases as the degree of compression is increased, and it is therefore

possible to reach a point at which the temperature will be high enough to cause the spontaneous ignition of the fuel. We should then have an approximation to the instantaneous addition of heat called for in the idealized cycle. This, for reasons given above, is undesirable in the practical engine, and conditions must be so chosen that anything approaching this instantaneous ignition is avoided even under the most rigorous conditions of operation. A definite measure of control must be retained over both the moment of ignition and the rate of burning, and this cannot be done if ignition by the heat of compression is allowed to take place. When the fuel and air are compressed together it is therefore necessary to limit the compression ratio to a figure which will ensure that control over the rate of burning is not lost at any moment during the process of combustion.

This limitation of the compression ratio limits the expansion ratio to the same figure, and therefore imposes a limit to the maximum efficiency of the cycle. The maximum compression ratio which can be employed depends upon a number of factors, the chief of which is the fuel itself. With ordinary commercial hydrocarbon fuels this maximum may be placed at about 6 : 1, which corresponds to an air standard efficiency of 52 per cent. Special fuels for aircraft engines allow figures of the order of 8 or 9 : 1, corresponding to an air standard efficiency of nearly 60 per cent.

It is not our purpose, however, to discuss carburettor engines, and the foregoing remarks have been introduced to explain the reason for adopting the device of injecting the fuel towards the end of compression, namely, to avoid the limitation in efficiency which is imposed upon us when fuel and air are introduced simultaneously.

3. Fuel and Air introduced Independently.

To avoid the difficulty just discussed the air is introduced and compressed alone, and the fuel is then introduced towards the end of compression, just before combustion should begin. By this means no limitation is imposed by combustion problems to the *upper* limit to which the compression can be carried, and we are free to use any value for compression ratio we may deem advantageous or desirable. Advantage is taken of the high compression temperature to provide for the ignition of the fuel and thus dispense with any special apparatus for this function. This feature, which gives the engine its name, imposes a *lower* limit to the compression ratio we may employ. This, however, is not a hard and fast figure and varies with the size of engine, type of combustion chamber, and nature of the fuels upon which the engine is intended to run. Speaking generally, the ratio best suited to any given set of conditions is ample to ensure satisfactory ignition under running conditions; when there is any difficulty in starting up

from cold the provision of a temporary source of additional heat serves to overcome any trouble of this kind.

Theoretically, the injection of the fuel during the combustion period enables us to control the rate of burning, and thus regulates the pressures within any required limits. In practice, however, it does not work out quite like this, as there are certain limitations both on the combustion side and also on the part of the injection equipment which limit the amount of control which it is possible to exert.

The withholding of the fuel until combustion should begin introduces problems of its own which have to be solved if satisfactory results may be obtained. The chief of these has been the bringing together of the fuel and air in such a way that the combustion may take place rapidly and completely and without shock in the short space of time available in an engine running at high speeds.

4. Combustion in the Compression-Ignition Engine.

The method by which the fuel is introduced into the engine results in combustion taking place under conditions vastly different from those which exist in a gas or petrol engine. In the latter, combustion takes place in a substantially homogeneous mixture of fuel and air, with the fuel in a gaseous or vaporized condition. Both the active reagents of combustion are of molecular dimensions, or very nearly so, and, provided that the distribution of the fuel is even approximately uniform, the fuel and air are in the best possible form for rapid and complete combustion. Even so, as was demonstrated by Dugald Clerk's classic experiment, when before firing the charge in a gas engine he arranged to compress and re-expand it repeatedly until the induction-induced turbulence had died out, the rate at which combustion occurs is too slow for even moderate engine speeds, unless it is assisted mechanically by the turbulent state of the gases in the cylinder. Since fuel and air are thoroughly mixed together, the combustion of one portion of the charge in no way interferes with that of the remainder; on the contrary, the increase in pressure and temperature which is produced by the combustion of the first part consumed results in an increasing rate of combustion as the process proceeds and may, under certain conditions, result in an undesirably rapid rate of burning during the final stages.

In the compression-ignition engine the fuel and air remain separated until combustion should start, and the fuel is then sprayed in a finely divided state into the highly heated mass of compressed air. The fuel and the oxygen have then to seek one another out, ignite and complete their reactions all in an extremely short space of time.

It may be emphasized here that the fuel does not, as is so frequently supposed, have first to be completely vaporized and mixed with the air before being ignited by the heat of compression. Such a sequence

of events would entail the loss of the very function which the principle of fuel injection is introduced to perform, namely, control over the time and rate of combustion. A little reflection will show that a mixture of fuel vapour and air ignited by the heat of adiabatic compression would ignite more or less instantaneously throughout its whole bulk; no control, therefore, could be maintained over either the exact moment at which ignition takes place or the rate at which the resulting combustion proceeds. If complete vaporization and mixing took place, except for the fact that we could ensure that ignition did not occur before some arbitrary point in the cycle was reached, we should be very little, if any, better off by injecting the fuel than by introducing it along with the air. If a satisfactory degree of control over combustion is to be maintained the vaporization of the fuel must as far as possible be avoided.

5. Ricardo's Three Stages of Combustion.

Ricardo has described combustion as taking place in three stages, as illustrated in the pressure-time diagram shown in fig. 40. Stage 1,

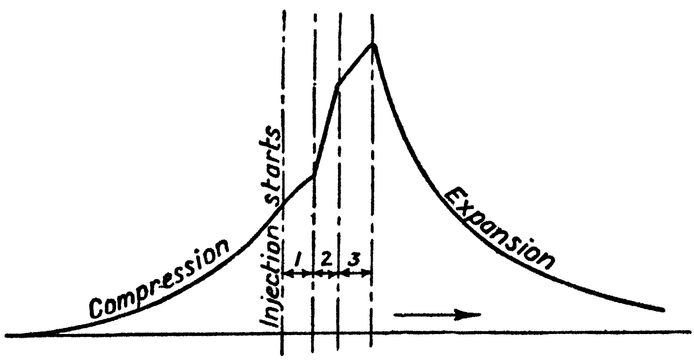


Fig. 40

usually termed the "delay period", is the interval of time which elapses between the commencement of injection and the first rise in pressure above the normal compression pressure. During this period fuel is being delivered into the cylinder and combustion is either non-existent or proceeds at such a low rate that the quantity of heat liberated is insufficient to produce a measureable deviation from the compression curve. Stage 2 commences at the moment when the pressure rises suddenly above the compression pressure and combustion, having started, spreads at a rapid rate through the fuel already present in the combustion chamber. This combustion is more or less uncontrolled, and the rate at which it takes place will depend chiefly upon the quantity

and condition of the fuel. A more or less complete state of inflammation having been reached by the fuel present, Stage 3 is reached, during which the fuel ignites almost as soon as it issues from the nozzle and is therefore more or less under the control of the fuel pump.

6. The Combustion of an Individual Fuel Particle.

In order to obtain a clear picture of what happens in the combustion chamber it will be of advantage to consider the career of a single fuel particle. The particle arrives in the combustion chamber endowed with a certain velocity which starts it travelling across the chamber through the compressed and heated air. On arrival it is at a comparatively low temperature and at once begins to absorb heat from the air, until a point is reached at which vapour will be formed. This vapour will tend to produce an envelope around the drop and, if the temperature of the air is sufficiently great, the envelope will ignite almost as it forms and the particle will therefore burn directly from the surface of the liquid and will go flying across the combustion chamber like a miniature meteorite.

The motion of the particle through the air will tear away the products of combustion and bring a steady stream of fresh oxygen particles into contact with the surface of the liquid, causing rapid combustion, and the particle will get progressively smaller until it is entirely consumed.

This represents the more or less ideal state of affairs wherein ignition takes place with a minimum of delay. The particle, though small, does not necessarily have to reach the ignition temperature throughout its mass. It is sufficient to bring its outer surface to the ignition point; once this is done, combustion will proceed from the surface of the liquid drop exactly as it does from the surface of a particle of coal-dust in a pulverized-coal-burning installation. In actual fact, apart from the difference in density of the atmosphere and the fact that combustion is intermittent in the case of the compression-ignition engine and continuous in the case of a pulverized-coal-burning installation, the process of combustion is essentially the same in both.

The temperature and density of the air within the combustion chamber are such as to tend to minimize the time required by the fuel to reach its ignition point, while the pressure is such as to tend to check the formation of vapour, so that under suitable conditions combustion can be promoted quickly and continued rapidly.

If, however, for any reason ignition does not take place with sufficient rapidity, the vapour envelope will grow and quickly reach proportions which will result in vapour molecules being torn off to mix with the air and form a combustible mixture to be ignited later. Should any appreciable quantity of vapour be formed, ignition when it takes place will spread through the vapour-air mixture with extreme

rapidity, causing a sudden increase in pressure and producing what has been called " Diesel Knock ".

This point is very clearly brought out in Report 435 of the National Advisory Committee for Aeronautics, Washington, D.C., where it is shown that vaporization can, and actually does, take place under certain circumstances even in the extremely short time available in a high-speed engine. It was found that whereas vaporization does not influence the ignition temperature it does influence the course of combustion, and any appreciable amount of vaporization results in violent combustion and roughness. From these experiments it was found that with a certain fuel and with an injection timing 20° early, complete vaporization took place when the engine was running at 1500 r.p.m. in a time interval of only ·0044 sec. (40°), the jacket temperature being 200° F. The importance of a short delay period if vaporization to any appreciable extent is to be avoided is thus made manifest. The experiment also emphasizes the importance of the boiling range of the fuel, a high boiling range reducing the amount of vaporization and vice versa.

The conclusions drawn in this Report are, in part, as follows:

1. " The percentage of fuel evaporated before ignition is a function of the boiling temperature, or temperatures, of the fuel, the injection advance angle, the engine temperature and the engine speed."

2. " Unless the combustion is started before the fuel vapours have diffused throughout the chamber the combustion will start almost simultaneously, with consequent combustion knock."

This latter conclusion is important, because if the vaporized fuel remains in a cloud and is not dispersed combustion can proceed only as oxygen makes contact with the outside of the cloud and cannot therefore proceed with very great violence; but, on the other hand, it is not likely to proceed with sufficient rapidity to give satisfactory operation in the engine. With the high rate of air movement necessary for a high-speed engine the chance of the vaporized fuel remaining in a cloud is remote, and the paramount necessity of reducing the delay period to a minimum and thus reducing vaporization to a minimum will be obvious. Once ignition has taken place the great increase in temperature ensures that no further accumulation of vapour can take place, and at the same time reduces to a minimum the time interval between the arrival of further fuel particles and their ignition. After the initial disturbance, therefore, the combustion proceeds in a quiet and orderly manner.

Those accustomed to the comparatively narrow range of " explosive mixtures " as these are understood in gas and petrol engine work may find it a little difficult to understand how a mixture which at most must contain a quantity of fuel far less than the minimum required

for an inflammable mixture, to say nothing of an "explosive mixture", can produce the pronounced knock which is sometimes heard. The explanation is that a mixture far too weak to be ignited by an electric spark or a flame may be strong enough to ignite with explosive violence when raised to its ignition point by adiabatic compression which brings practically every part of the mixture to its ignition point simultaneously. The degree of violence will then depend solely upon the quantity of energy liberated.

The foregoing description is intended to give, from the engineering standpoint, a general idea of what takes place during the delay period, and although it does not take into account any of the complex chemical reactions which may occur during the interval immediately preceding ignition, it will be sufficiently accurate for general purposes.

7. The Delay Period and the Working Cycle.

The cycle of operation upon which the high-speed engine works is settled by the delay period. The existence of the delay period makes inevitable a proportion of constant volume burning at the beginning of combustion and therefore precludes entirely any possibility of working on the true constant pressure cycle or even an approximation to it.

This does not mean that the delay period does not exist in, or has no influence upon, the slow-speed engine, but the relationship which the delay period, measured in time, bears to the duration of the injection period is so much greater in the case of the high-speed engine that its influence is increased out of all proportion. Even low-speed engines when fitted with mechanical injection will receive sufficient fuel during the delay period to produce a definite period of constant volume burning and a pressure rise above the compression pressure sufficient to make it impossible to consider them as working on the constant pressure cycle.

In the high-speed engine the delay period has been reduced to as little as ·001 sec., and even less, values as low as ·0006 sec. not being unknown; but, short as this time is, it is nevertheless a quite big fraction of the total injection period when speeds of the order of 1500 r.p.m. and upwards are considered. Under full load the injection period normally occupies some 20° to 24° of crankshaft rotation and at 1500 r.p.m. 20° requires only ·00222 sec.; a delay of ·001 sec. therefore represents nearly half the total full-load injection period, and at lighter loads the proportion becomes even greater. The fact that the delay period may occupy nearly half the total injection period does not necessarily mean that a corresponding proportion of the total fuel has been delivered before ignition takes place; the fuel spray requires time to build up to its full discharge rate, so that the proportion of fuel delivered is somewhat less.

The actual quantity of fuel delivered during the delay period is

not easy to determine accurately, but it is a fairly large proportion of the total charge, and although only a part of it may be consumed in the very rapid period of combustion which occurs during the few degrees of crankshaft movement immediately following ignition, there is sufficient fuel involved to ensure that the initial stages of combustion take place under what are substantially constant volume conditions, and with a fairly high pressure increase.

The quantity of fuel involved in the period of constant volume combustion will depend upon the length of the delay period, the average rate at which the fuel has been delivered during the delay, and the proportion of the fuel which is in such a condition that it can be burned almost instantaneously. The latter will depend to a great extent upon the fineness of the spray and the amount, if any, of vaporization which has taken place.

The normal method of regulating the fuel delivery to suit the load upon the engine is to vary the duration of injection rather than to change the rate at which injection takes place. The effect of any alteration in load will be to vary the amount of fuel which is received after the delay period. This will be true no matter whether the change in pump delivery is made at the beginning or the end of the delivery, because the delay itself always takes place at the beginning of the injection period.

With any given engine running at a given speed a reduction in load will not result in a reduction in the quantity of fuel which is burned at constant volume, but only in that which is delivered after the constant volume combustion has taken place. No change in the amount of constant volume combustion will take place until the total quantity of fuel is reduced to something less than that which is normally delivered during the delay period, i.e. until the length of the injection period becomes less than that of the delay period. This statement requires a little amplification; any increase in the delay period such as often occurs with a reduction in load will, of course, result in an *increase* in the quantity of fuel available for constant volume combustion.

It was shown (pp. 15, 67) that the efficiency of the mixed cycle depends upon the ratio of the fuel burned at constant pressure and that burned at constant volume, the maximum efficiency for any given expansion ratio being obtained when the whole of the fuel is burned at constant volume. It will be clear, therefore, that the efficiency of the engine will increase as the fuel charge is reduced, because the cycle is gradually changed from the mixed cycle to the constant volume cycle. This will be shown by the diagram (fig. 41), which gives the conventional mixed cycle diagram; *ab* represents the constant volume burning, and *bc* the constant pressure burning. As the quantity of fuel is reduced the amount of constant pressure burning is reduced

also, as indicated by the dotted lines, and the efficiency increases until finally *bc* becomes zero, combustion takes place wholly at constant volume, and the efficiency reaches that of the constant volume cycle. Any further reduction in the quantity of fuel results in a reduction in the quantity of fuel burned at constant volume, and theoretically results in no further improvement in the efficiency of the cycle. Under practical conditions the efficiency will, of course, be improved owing to a reduction in the maximum temperature whether this occurs at constant pressure or constant volume, but an improvement from this cause is quite distinct from, although taking place simultaneously with, that occasioned by the reduction in the extent of the constant pressure combustion.

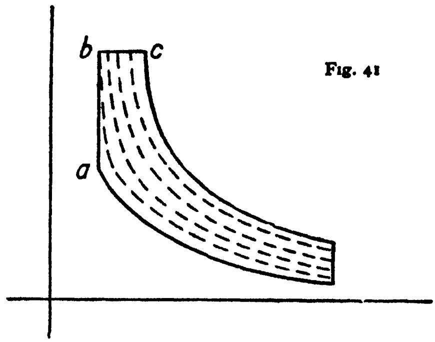


Fig. 41

The delay period thus exerts a very definite influence upon the cycle of operation. It absolutely prevents a high-speed engine from being operated upon the constant pressure cycle, or its practical equivalent the Diesel cycle, and at the same time it settles whether the engine shall work upon the mixed cycle or the constant volume cycle and, in the event of the former, settles the proportions of the fuel burned under the two types of combustion.

8. The Delay Period and Engine Behaviour.

The general behaviour of the engine is very largely dependent upon the way in which the combustion is divided up into Ricardo's three stages. It will be readily understood that a long Stage 1 will be likely to be disadvantageous on account of the large quantity of fuel which will be burned under conditions which do not admit of adequate control, and the smoothness of operation of the engine will be affected adversely on account of the rapid rate of pressure rise during Stage 2. On the other hand, other things being equal, the larger the quantity of fuel already within the combustion chamber when ignition takes place the nearer will the combustion be to taking place under constant volume conditions and the more efficient will the engine tend to be. This explains the somewhat improved fuel consumption which is frequently obtained when operating with a fuel which gives a long delay and rough running.

The three stages are not, of course, as clearly defined as is indicated in fig. 40, and it is frequently somewhat difficult to determine the

demarcation of each. Stage 1, as would be expected, is by far the most clearly defined. Its commencement is given by the lifting of the injector nozzle valve and its termination by the point at which the pressure rise leaves the compression curve. This latter is not always very sharply defined, especially if the ignition takes place at a point at which the compression curve is rising rather steeply, and the combustion pressure rise leaves the compression curve nearly at a tangent. Generally speaking, however, it is not very difficult to say pretty accurately when Stage 1 ends and Stage 2 begins. It is the demarcation of Stage 3 which offers the greatest difficulty, and it must be remembered that Stage 3 may not actually exist in some instances, especially under light-load conditions, because the injection period may have ceased before Stage 2 has ended, and, in extreme cases, even before it has been begun.

The trend of the pressure during Stage 3, as indicated in fig. 40, must not be taken too seriously. Just what changes take place in the pressure during this period will depend upon a number of factors, such as the rate of injection, the rate at which the fuel burns (and this itself depends upon a number of factors), at what point in the cycle Stage 3 begins and for how long it continues, i.e. the changes in volume due to piston movement during the period, and the gain or loss in heat which takes place.

Combustion does not cease with the termination of injection. Injection ends when the last particle of fuel issues from the nozzle, but this last particle, like all those which preceded it, requires time to secure the oxygen necessary for its complete combustion and, because of the relative scarcity of oxygen, will usually require a longer time than its forerunners. Combustion thus goes on after injection has ceased and continues until late-coming particles are either satisfied or finally give up their search for oxygen. There is thus, in reality, a further stage which is to some extent independent of the other three and which actually overlaps Stages 2 and 3, because the combustion of the fuel delivered during Stage 1 will rarely be completed by the time Stage 2 has ended, while at the same time fuel is being received throughout the whole of Stage 2. Except for Stage 1, which is in actual fact pretty sharply defined even in those cases where its termination is not too well marked on the indicator diagram, the three stages form a convenient method for describing a sequence of events rather than three distinct and well-defined phases which follow one another in regular order.

It is sometimes supposed that combustion is completed at the point at which the maximum pressure is reached. This may be true in certain cases, but is very far from being generally true. As has already been pointed out, the point at which the maximum pressure is reached is nothing more than the point in the cycle at which the

increase in pressure resulting from the rise in temperature produced by combustion is balanced by the fall in pressure consequent upon a loss of heat and the increase in volume occasioned by the outward movement of the piston.

The delay period, or ignition lag as it is also called, thus plays a very important part in the operation of the engine. It is no exaggeration to say that the delay period really determines whether or not the engine gives satisfaction in service. The delay governs the smoothness of running and to a very large extent the freedom of the engine from mechanical troubles. The amount of uncontrolled burning which takes place in Stage 2 depends directly upon the quantity of fuel admitted during the delay period. In extreme cases it is possible for the whole of the fuel to be received during the delay period, and this may occur even under full-load conditions. Little or no control can then be exerted over the rate of burning; the engine will be prone to very rough running, and at the same time the high rate of pressure rise will result in the structure being subjected to severe shocks. One of the chief problems is to keep the delay period as short as possible, so that the minimum quantity of fuel is present in the cylinder when combustion commences. The initial rate of pressure rise will thus be kept moderate, and Stage 3 will reach a maximum; we shall then be able to exert a real measure of control over the combustion period.

The average rate at which fuel has been delivered during Stage 1 is, to some extent, under the control of the designer, although not entirely so because of the compressibility of the fuel and spring in the injection system, and also because considerations connected with the all-round efficiency of the engine demand a rate of injection such that it is not possible to keep down the rate during Stage 1 as much as might perhaps be desired.

The factors which govern the duration of the delay period are the ignition point of the fuel, or rather, the ignition quality of the fuel (see below); the compression temperature; the density of the compressed air, and to a limited extent, the fineness of the spray.

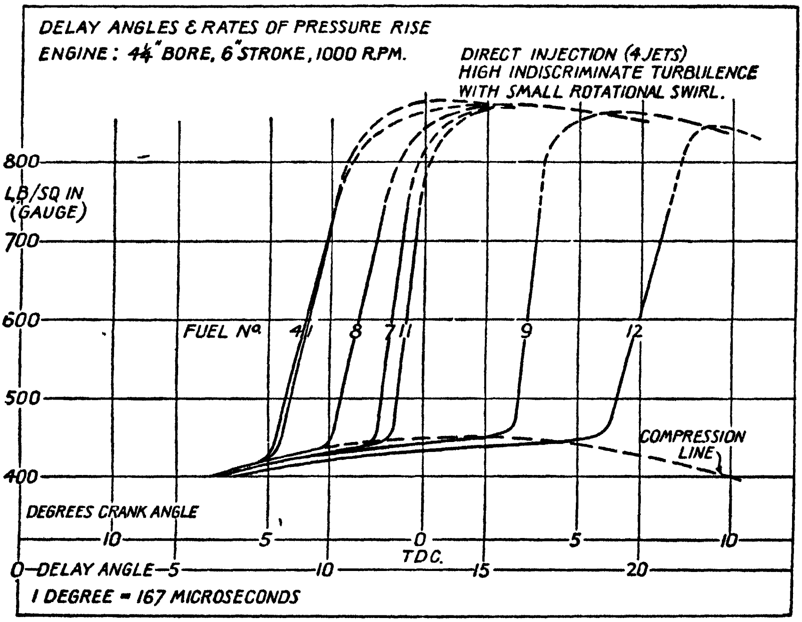
9. The Ignition Quality of the Fuel.

The ignition quality of fuels differs widely and depends upon the nature of the crude oil from which the fuel was distilled. The term "ignition quality" is used to denote the relative suitability of oil for use as a fuel in a compression-ignition engine, and includes all the properties which go to influence this quality. It is not as yet definitely agreed what the factors are which govern the ignition quality, but it appears that ultimately the self-ignition point of the fuel and its molecular structure are the chief factors.

All commercial fuel oils are a mixture of different members of the various groups into which the mineral hydrocarbons are divided, such

as the paraffins, napthenes and aromatics. The molecular structures of these different groups differ widely, and at the same time the ignition temperatures of the different members of the same group differ from one another and from those of the corresponding members of the other groups. The ignition point of any one fuel thus depends upon the proportions of the different hydrocarbons which may be present.

It might be thought that the ignition temperature of a fuel would



By courtesy of N. E. Coast Inst. Eng. and Ship.

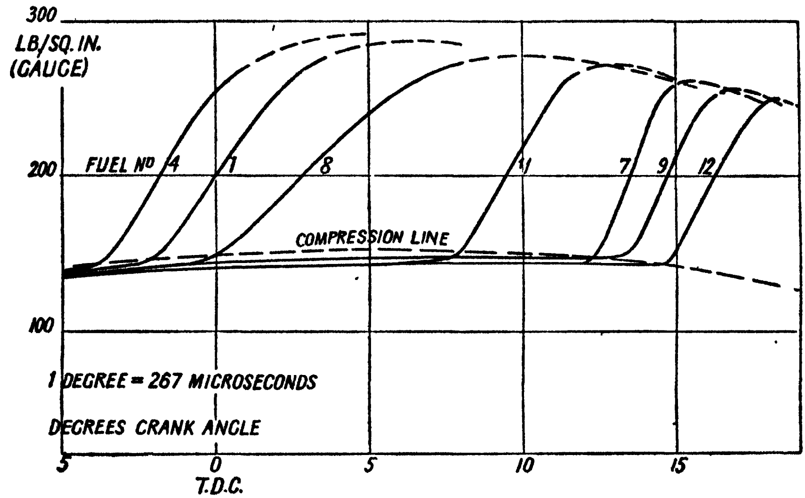
Fig. 42.—Variation in delay with different fuels. Open combustion chamber engine

be that of the lowest member it contained. This, however, is not the case; the ignition point depends upon the proportions of the various members it contains. Thus a mixture in equal proportion of two fuels which had ignition points of 350° C. and 435° C. in a given apparatus was found to have an ignition point of 380° C., while a mixture of 60 per cent of that having the lowest ignition point and 40 per cent of the other had an ignition point of 375° C.

The difference in delay period which exists with different fuels is well illustrated by fig. 42, which is taken from a paper by Le Mesurier and Stansfield.* This shows the extent of the delay period produced

* *N.E. Coast Inst. Eng. and Ship.* (1932).

by seven different fuels in the same engine. It will be observed how the rate at which the pressure rises increases with the delay, and also how the height to which the pressure rises before the rate of rise slows down increases as the delay increases. In the case of the fuel showing the maximum delay, fuel 12, the rate of pressure rise is seen to have fallen off and to be slightly less than that of the two fuels having the shortest delay. This can be explained by the fact that whereas in all the other instances ignition takes place before, or only very slightly



By courtesy of N. E. Coast Inst. Eng. and Ship.

Fig. 43.—Variation in delay with different fuels

Note: Fuels 4 and 11 are not strictly comparable with the others, owing to high viscosity advancing the injection point.

Engine: Hot-bulb two-stroke 6-in. bore, 5¾-in. stroke, 625 r.p.m.; single open type centrifugal spray, moderate indiscriminate turbulence.

after, top dead centre, the delay with fuel 12 is so great that ignition does not take place until appreciably after top dead centre; it therefore takes place while the volume is increasing rapidly and this reduces the rate of pressure rise.

In obtaining these diagrams, the brake mean pressure, the speed, water jacket temperature, air inlet temperature and injection timing were all kept constant, so that the results shown are solely the effects of using different fuels. The delay period varies from 8° to 19°, amounting to from ·00132 to ·00318 sec.

A similar series of results is shown in fig. 43, from the same paper. These are taken from a hot-bulb engine, a type which for various reasons has a much greater delay than is usual with high-speed compression-ignition engines. In this instance the two fuels showing the

shortest delay, Nos. 1 and 4, show a somewhat higher rate of pressure rise than fuel No. 8, this being due to the fact that the short delay period of these two fuels results in the injection timing being too early.

The ignition point of a fuel is not a definite point on the temperature scale, but varies considerably according to the conditions under which the measurement is made. Experiments for determining the ignition point have been carried out in numerous different ways, but no satisfactory agreement between different methods has been obtained, and in many instances different methods do not even place a group of fuels in the same order as regards ignition point.

The existence of the delay period makes it a difficult matter to decide just what temperature is to be considered as being the ignition point of a fuel. With all fuels the delay period decreases as the temperature of the apparatus to which the fuel is introduced increases. In some fuels there is a fairly well defined temperature below which ignition will not occur, but above which ignition will take place after only a short delay. Others, however, have a wide belt of temperature over which the delay steadily increases with a decrease in temperature, until just before the temperature at which the fuel fails to ignite the delay has reached a period amounting to many seconds.

A just perceptible pause before ignition takes place in a test apparatus involves a lapse of time which when considered in terms of high-speed engine operation is one of very material duration. A pause before ignition which would be just noticeable to an alert observer is more than sufficient for several complete cycles of a fast-running engine.

The ignition temperature of the fuel is greatly influenced by the density of the atmosphere under which the measurement is made, and decreases with an increase in density. Considered from a chemical standpoint, combustion is an oxidation process and the process may take place either gradually or with exceeding rapidity or with an infinite variety of speeds between, the quantity of heat evolved in the process being the same for all. The rate at which oxidation takes place is increased by an increase in temperature and also by an increase in pressure, i.e. density. The effect of the increase in temperature is to increase the chemical activity, while an increase in density increases the intimacy of the contact between the two reagents and thus facilitates their combination.

At low rates of oxidation the rate of evolution of heat is so slow that no appreciable temperature rise may be detected, because the heat can escape as fast as it is evolved. As the rate of evolution increases, however, the temperature begins to rise; this rise in temperature assists oxidation, and thus tends to increase the temperature yet further. The two thus react upon one another with the result that finally the rate of evolution of heat becomes so great that a rapid rise

in temperature takes place accompanied by the inflammation of the material, producing the phenomenon known as combustion.

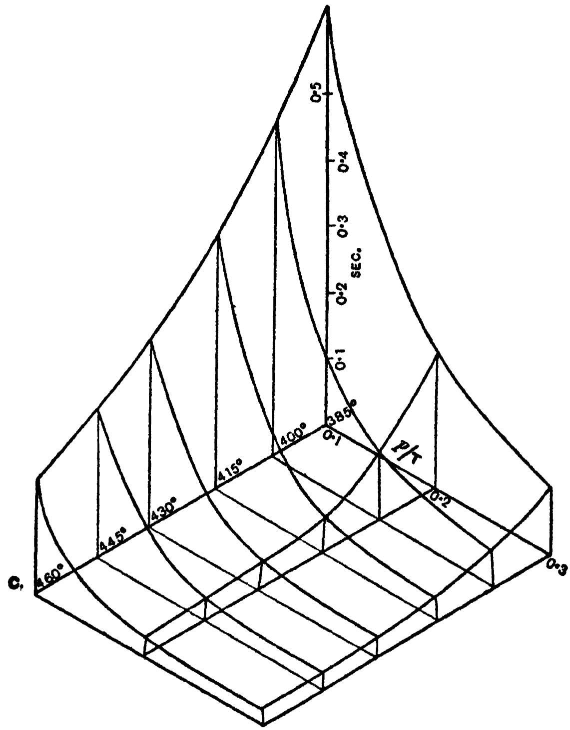
The intimacy of contact between the fuel and the oxygen molecules of the air depends upon the number of the molecules present per unit volume of air, and this will be increased by concentrating a greater weight of air in a given volume. The weight of a given volume of any gas depends not only upon the pressure but also upon the temperature, and it is thus the *density* of the air rather than the pressure alone which governs the rate at which oxidation takes place. At any given temperature the rate of oxidation increases as the density of the air is increased; an increase in density therefore causes a decrease in the temperature at which the rate of heat evolution is such that combustion follows. That is, an increase in density of the air results in a decrease in the ignition temperature of the fuel and vice versa.

Investigations into the influence of density upon ignition temperature have been carried out by a number of experimenters, notably by Bird at Cambridge University. In Bird's experiments the fuel was injected at a pressure of 2000 lb./sq. in. into a bomb containing air under pressure and heated to the required temperature. The interval of time between the moment of injection and that of ignition, as shown by a sudden increase in pressure, was recorded by a rotating drum indicator. The results of some of these experiments are shown in fig. 44, which is taken from one of his papers.*

The figure is in the form of a three-ordinate diagram which gives the relationship between delay period, density and temperature, the density being expressed in terms of absolute pressure divided by absolute temperature. This diagram shows the very marked reduction in the delay period which follows upon an increase in density. It also shows that the influence of increase in density is a decreasing one, as is also that of temperature. Bird's experiments indicate further that with the fuel he used, one derived from a paraffinic base crude and, therefore, one of good ignition quality, a minimum delay of ·03 sec. is to be expected, his curves flattening out in such a way as to suggest that no further increase either in temperature or density will result in any further decrease in the delay period. A minimum delay of this order would render the high-speed engine an impossibility, and in actual fact Bird's minimum delay is from 25 to 40 times as great as has actually been found to occur in high-speed engines and five or six times as great as in the case of low-speed engines. The maximum temperature used in these experiments was 460° C., which is about the bottom end of the scale of temperatures found in high-speed engines, while his pressures also appear to have been low for high-speed engines. The fact that in his experiments he apparently reached the minimum

* *Proc. Inst. Mech. E.*, Nov., 1926.

value of delay suggests that no advantage is to be gained by making tests with greater temperatures and densities.



By courtesy of the Inst. of Mech. Eng.

Fig. 44.—Showing relation between delay period, density, and temperature, as determined by Bird

10. Compression Temperature and Delay.

The influence of temperature upon delay in an actual engine is shown in a striking manner by some experiments of Le Mesurier and Stansfield,* where the temperature of the air received by the engine and also that of the jacket water were maintained at a uniform temperature and varied over a range of temperature from 20° C. to 100° C. with the engine running at a speed of 1000 r.p.m.

Several fuels of differing ignition quality were tested, and all exhibit a progressive decrease in delay as the temperature is increased, as is shown in fig. 45. The fuels of good ignition quality are less affected than those of lower ignition quality, but all show the same trend, and all confirm that the effect of a given increase in temperature decreases

* *N. E. Coast Inst. Eng. and Ship.*, Feb., 1932.

as the temperature increases. The trend of the curves shows that an increase in temperature beyond the maximum used for this experiment would produce some reduction in the delay even in the case of the fuel having the best ignition quality, while with that having the lowest ignition quality the slope of the curve indicates that a very material reduction in delay could have been obtained by a further increase in temperature. In the case of the two best fuels the trend of the curve suggests that the delay would be reduced to about 5° for a temperature of 200° C., while that of the worst would not be much in excess of this figure.

Le Mesurier and Stansfield do not give any information as to the compression temperatures corresponding to the different induction

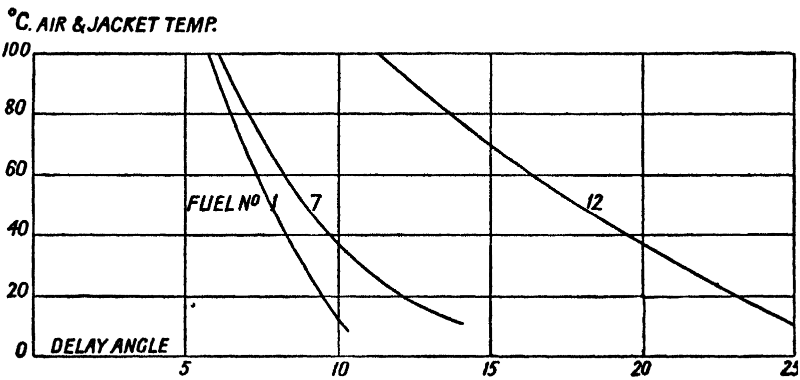


Fig. 45.—Influence of temperature upon delay. Open combustion chamber engine, 4½ in. bore, 6-in. stroke, 1000 r.p.m.; direct injection (4 jets)

and jacket temperatures, and in the absence of further data it is not possible to arrive at a very accurate determination. The compression ratio is, however, quoted as being 13 : 1 and the effective compression ratio may therefore be taken as about 12 : 1. We know that some increase in temperature of the air will take place during induction because the temperature of the hotter parts of the combustion chamber will be appreciably above that of the jacket water, while, in addition, there is the effect of the residual exhaust gases. This, however, is comparatively small.

All the available data from which the final suction temperatures can be ascertained have been obtained with air at normal atmospheric temperatures and with jacket temperatures materially above that of the air. Such data, therefore, are not directly applicable to the present case. It would appear, however, that the amount of heat picked up under the different temperatures in this experiment would be much the same for all, and because the air and the cooler parts of the metal with which the air is in contact are at the same temperature, it would

probably be somewhat less than the figure given in the discussion of volumetric efficiency (p. 88). If, therefore, we take a figure of 10° C. for the rise in air temperature during induction and assume that the residual exhaust gases increase the temperature by 4 per cent, it would seem that we are not very far from the truth and err, if anything, on the side of moderation. On those assumptions, and taking the value of n as 1·35 (this might quite likely increase somewhat at the higher temperatures) the compression temperatures shown in Table XV are obtained.

TABLE XV

Air and Jacket Temp.		Final Suction Temp.	Compression Temperature		Delay					
					Fuel 1		Fuel 7		Fuel 12	
°C.	deg. abs.	deg. abs.	deg. abs.	°C.	deg.	sec.	deg.	sec.	deg.	sec.
100	373	399	950	677	6	·001	6·2	·00103	11·3	·00188
80	353	378	897	624	6·5	·00108	7	·00117	13·8	·0023
60	333	357	850	577	7·3	·00122	8·1	·00135	16·3	·00272
40	313	336	800	527	8·3	·00138	9·6	·0016	19·4	·00323
20	293	316	752	479	8·5	·00142	12·2	·00203	22·9	·00382

At 200° C. the compression temperature would have increased to 907° C.

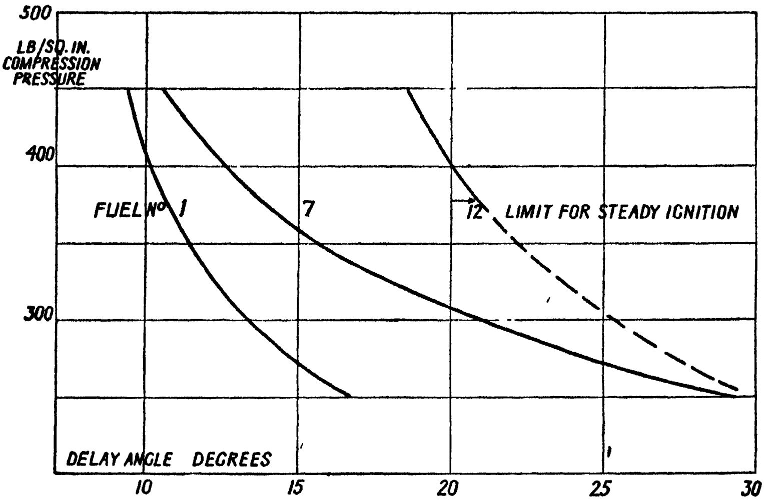
The inference from these figures is that under engine conditions the effect of temperature is much more marked than in bomb experiments such as Bird's, although both indicate a tendency towards a minimum value for the delay which will not be reduced further even if the temperature is carried to extreme limits.

11. Air Density and Delay.

The influence of density upon delay in the engine has been investigated both by Le Mesurier and Stansfield (*loc. cit.*) and Boerlage and Broeze. They used similar methods, throttling the air intake and thus reducing the density of the atmosphere received by the engine. The former, however, carried out their experiments with a constant temperature of the air as measured at the inlet valve, and the results which they obtained when using the same three fuels as those used for the temperature tests just cited are given in fig. 46. The results are, of course, for a *decrease* in density such as would result from operation at high altitudes or with a reduced compression ratio. The effects of an increase in density can, however, be visualized from the curves. As with temperature, so with density; the effect of a given increase in density is a decreasing one, and a minimum value for delay is ultimately reached, which thereafter remains uninfluenced by any further

increase in density. It will be observed that in the case of fuel No. 12 steady ignition could not be obtained when the compression pressure was reduced below about 370 lb./sq. in., while the other two fuels, which had a much shorter delay period, gave steady ignition, but with a much increased delay period, down to a pressure of 250 lb./sq. in., which appears to be the limit of the experiment.

This is an important feature and was taken advantage of by Boerlage and Broeze to determine the relative ignition qualities of different fuels. Their investigations into ignition quality have thrown con-



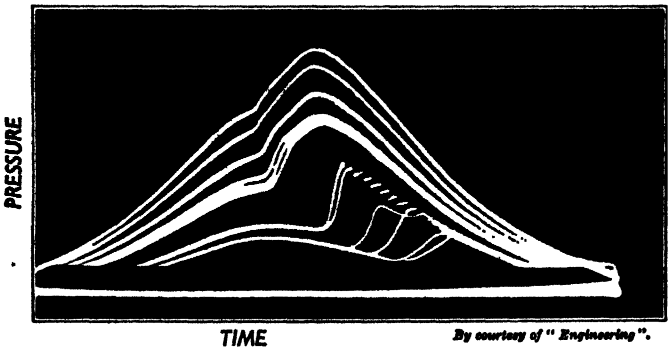
By courtesy of N. E. Coast Inst. Eng. and Ship.

Fig. 46.—Influence of temperature upon delay. Open combustion chamber engine: 4½-in. bore, 6-in. stroke; direct injection (4 jets). Inlet air temperature between throttle valve and inlet valve 30° C.; jacket, 60° C.; speed, 1000 r.p.m.

siderable light on the influence of density upon delay period under engine conditions. They have shown that a reduction in air density involves a very material increase in the delay period of all fuels, together with an increase in the rate of pressure rise and in the tendency towards roughness in running. In other words, the combustion knock and delay period are associated, an increase in the latter involving an increase in the former and vice versa. A fuel which normally gives smooth burning may be made to knock by the simple process of reducing the density of the air within the cylinder. Conversely, a fuel which gives rough running may be made to burn more smoothly by an increase in air density, i.e. by supercharging. The influence of the compression ratio upon the smoothness of running is thus emphasized.

Boerlage and Broeze's method was the same as that of Le Mesurier

and Stansfield, except that no attempt was made to ensure a constant temperature of the air at the inlet valve. The method of the former, therefore, is somewhat more easily applied. The engine is allowed to reach its equilibrium temperature when running on a fairly small load, about one-quarter to one-third of its full-load output, and the air intake is then progressively throttled during a few cycles while a series of indicator diagrams is taken on a single card. Apart from the throttling no change is made, and the whole test is carried out as quickly as possible in order that there shall be no time for temperature changes of any magnitude to occur. The experiment is carried out at a reduced load in order that combustion conditions may be interfered with as little as possible. A typical series of such diagrams taken from their



By courtesy of "Engineering".

Fig. 47.—Pressure-time diagram showing increased delay with reduction in compression pressure

published work is shown in fig. 47, and indicates very clearly how the delay period increases steadily as the compression pressure, and therefore the density, decreases.

As the delay period increases the initial rate of pressure rise at first increases also, the amount of fuel present at the moment of ignition having been increased by the prolonged delay. A roughness in running makes itself felt if the fuel is a smooth-running one, or an increase in roughness if the fuel is already prone to roughness. As the density is decreased the delay increases at an increasing rate, as does also the roughness, until, ultimately, a point is reached at which the re-expansion of the air caused by the outward movement of the piston makes itself felt, the rate of pressure rise is reduced, and an *improvement* in smoothness follows. Reducing the density still further causes a yet greater delay, and finally, misfiring occurs.

An increase in density, such as may be produced by supercharging, produces the opposite effect, the delay period shortening as the density increases, with an improvement in smoothness. The results of some experiments in this direction, carried out by the author, are shown in

fig. 48. The engine used was a single-cylinder experimental engine fitted with a Ricardo Comet Mark I head, and also with an open combustion chamber head. The pressures were varied from —14 in. of mercury to +15 in., but with the set-up employed for the Comet head a marked ramming effect was produced by the length of pipe used to connect the engine to the air tank, and a pressure in the tank equal to normal atmospheric pressure was actually equivalent to a pressure of +2 in. of mercury in this experiment. As will be seen, the delay period increases steadily as the induction pressure is reduced but decreases with an increase in pressure, falling to a minimum value of

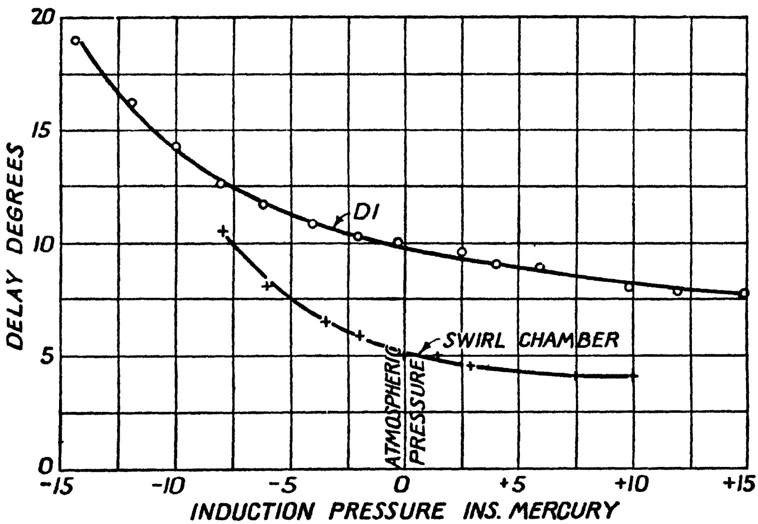


Fig. 48.—Influence of induction pressure upon delay

about 4°, equivalent to ·00044 sec., in the case of the Comet chamber. The indicator diagrams from which the information was obtained were taken by a Farnboro' indicator. At induction pressures above 7½ in. of mercury it was not possible to determine the length of the delay accurately, because the pressures swept up to the maximum in a fair curve; the readings between 4 in. and 7½ in., however, indicated that the minimum value had been reached. The readings for the open chamber were obtained by the use of rate of change of pressure diagrams, using a Sunbury indicator, which allows the delay to be determined with considerable accuracy. The trend is the same for both types of chamber, but in every case the delay is considerably greater with the open chamber.

These various experiments show that a fuel which normally gives smooth running in a certain engine can be made to knock merely by

reducing the density at the moment of injection, the effect of such a reduction being to produce an increase in the delay period. On the other hand, a fuel which gives rough running may be made to run more smoothly by increasing the density at the moment of injection, e.g. by supercharging or an increase in compression ratio,* either of which decreases the delay period. Such results confirm the statement that it is the delay period that governs the smoothness of the engine.

It must be admitted that changing the density in the manner used in these experiments introduces a certain amount of disturbance in other directions, and the effects produced may not therefore be solely those due to the change in density. The penetration and distribution of the fuel spray will undoubtedly be changed considerably; the fuel/air ratio is changed also, but by carrying out the experiment with a great excess of air any influence from this cause is minimized. The compression temperature will be changed somewhat, although theoretically, the compression ratio not having been altered, the temperature at the end of compression will remain the same. Such change in temperature as does take place will be caused by any reduction in air temperature which is produced by throttling, any difference in the quantity of heat picked up during induction, and the greater relative bulk of the residual exhaust gases, which will probably increase in temperature as the amount of throttling increases. The greater relative bulk of exhaust gases may of itself introduce a disturbing factor, because the quantity of inert gases contained therein will increase with the amount of throttling. Despite these doubtful factors, the experiments demonstrate in a most effective way the general effect of density upon the delay period or, perhaps one should say, ignition quality, of a fuel under engine conditions.

The conditions of temperature and density as they exist in the combustion chamber at the moment at which injection commences thus are clearly of the first importance, and have a very direct bearing upon the behaviour of the engine.

12. Temperature and Density in the Cylinder during the Delay Period.

The conditions which exist at the moment when the first particles of fuel enter the combustion chamber depend upon the instantaneous compression ratio, i.e. the compression ratio corresponding to the position of the piston at that particular instant, and it is the difference between these conditions and the minimum condition necessary to produce ignition which, to a large extent, governs the delay period.

Conditions such that ignition can occur are not obtained until the compression has been carried to a certain point. Obviously, therefore, if the fuel is introduced before the necessary conditions are reached

* An increase in compression ratio involves an increase in temperature also; its effect is therefore twofold.

the delay must be extended at the least to the extent of the period during which the conditions are less than the minimum necessary for ignition. This will be illustrated by fig. 49, which shows a compression diagram on a crank-angle base. Here the heavy full-line curve P represents the pressures corresponding to the different positions of the crank, while the broken curve ρ and the chain-line curve T represent respectively the corresponding values of density and temperature. The ignition temperatures of the fuel corresponding to the conditions of density and temperature are shown by a full line.

Starting at bottom dead centre, the conditions of density and pressure and the requirements of the fuel are such that ignition cannot

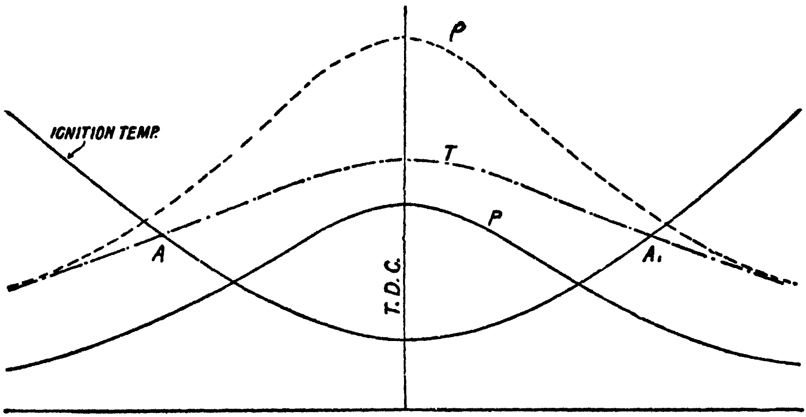


Fig. 49

possibly take place, but as the compression increases the conditions improve, the density and temperature increasing while the ignition temperature falls, until a point A is reached at which the temperature within the cylinder is equal to the ignition point of the fuel at the corresponding density. From this point onwards and until top dead centre is reached the conditions become more and more favourable towards the production of ignition, both on account of the increase in density and temperature, and also because of any corresponding reduction in ignition temperature. Once the top dead centre has been passed, however, the conditions deteriorate steadily, until beyond the point A′ the air temperature again falls below the ignition point of the fuel.

It follows, therefore, that ignition, if it is to take place at all, must do so between A and A′. Injection may, if so desired, be made to take place before A is reached, but under no circumstances can ignition take place before A is reached.

The duration of the delay will be some function of the average conditions which prevail between the injection point and the moment

of ignition. Should injection take place before A is reached it does not mean that ignition will take place at A. The actual point at which ignition takes place will depend upon the time the fuel requires to reach the necessary temperature. A is nothing more than the point in the cycle at which the minimum conditions for producing ignition are reached. It will be observed that the earlier in the cycle A is reached, the greater is the time available in which to produce ignition; not only is the available time increased, but also the margin of cylinder conditions above the minimum necessary for ignition. For any practical engine it is only the interval between A and top dead centre, or, at the most, only very shortly after top dead centre, that is available for producing ignition. Even in the case of engines running at speeds of 250 or 300 r.p.m., ignition cannot be allowed to occur more than a few degrees after top dead centre if satisfactory efficiency is to be attained, but for efficient working in high-speed engines it is essential for ignition to originate at a point which will enable the maximum pressure to be reached not later than about 15 deg. after top dead centre. This means that ignition must occur slightly before top dead centre is reached if an undesirably high rate of pressure rise is to be avoided. Under these conditions the injection must begin at such a point that, after allowing for the delay, ignition will occur at the desired moment.

It has already been stated that the delay is some function of the average conditions which prevail between the injection point and the moment of ignition, and from fig. 49 it will be clear that for a given delay, the later the injection the higher the average of the conditions. For present purposes the average conditions may be represented as the area between the ignition temperature curve and the air temperature curve contained between two vertical lines drawn through the injection point and the ignition point. If, therefore, the delay is dependent upon the average conditions which prevail during the delay, it follows that the lower the average conditions, the longer the delay. From this it may be inferred that even though injection does not begin until after the point A has been passed, the earlier the injection timing, the longer the delay. In practice, this is found to be the case; as the injection timing is advanced, the delay period increases. This increase is small at first, but gets more and more as the timing gets earlier and earlier. The same thing is true with a change in compression ratio. For a given injection timing the delay decreases as the compression ratio is increased. This means that the effect upon delay of a given change in injection timing is also influenced by the compression ratio of the engine.

The instantaneous compression ratio increases very rapidly as the piston approaches the inner limit of its stroke, and at the same time there is a steady decrease in the distance travelled by the piston for

a given angular movement of the crank. A comparatively small angular change in injection timing may therefore result in quite a large change

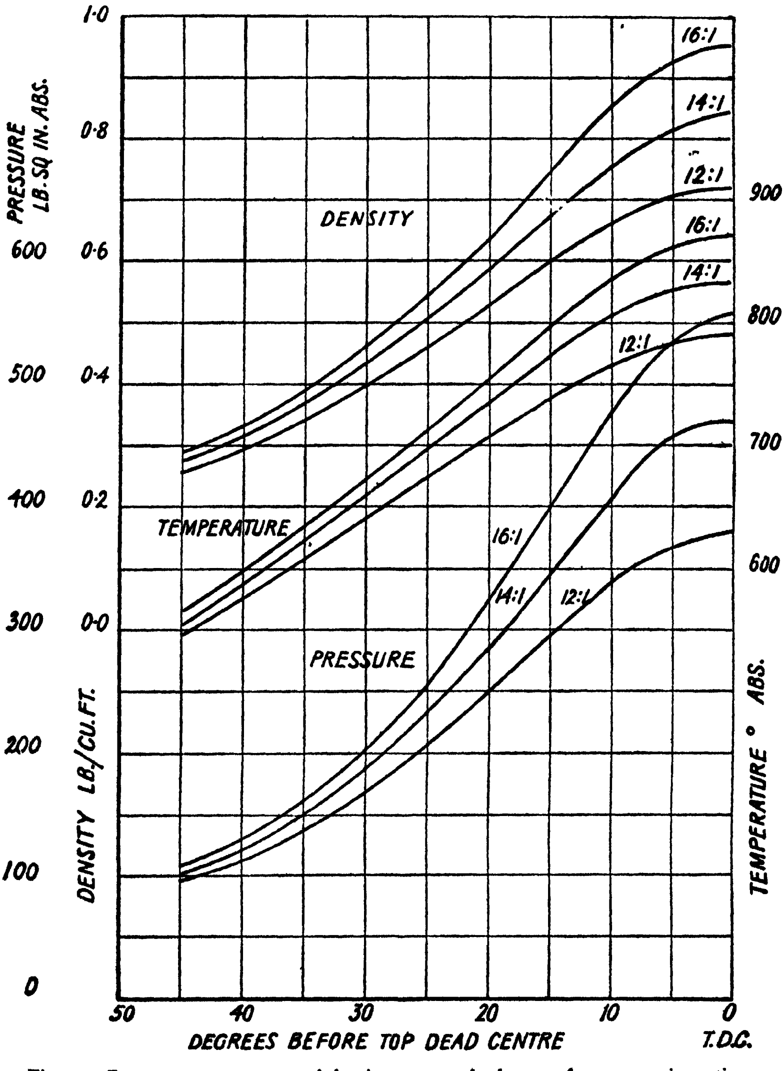


Fig. 50.—Pressure, temperature, and density near top dead centre for compression ratios of 12, 14 and 16 : 1

in the temperature and density conditions prevailing at the moment when injection commences. The extent of the changes is illustrated in fig. 50, which gives the pressures, temperatures and density for a

period of 45° before T.D.C. for compression ratios of 12, 14 and 16 : 1. In determining these values the effective compression ratio of the engine (see p. 90) was used, and the temperature at the beginning of compression was taken as being 340° abs., while the value of n was taken as 1·35.

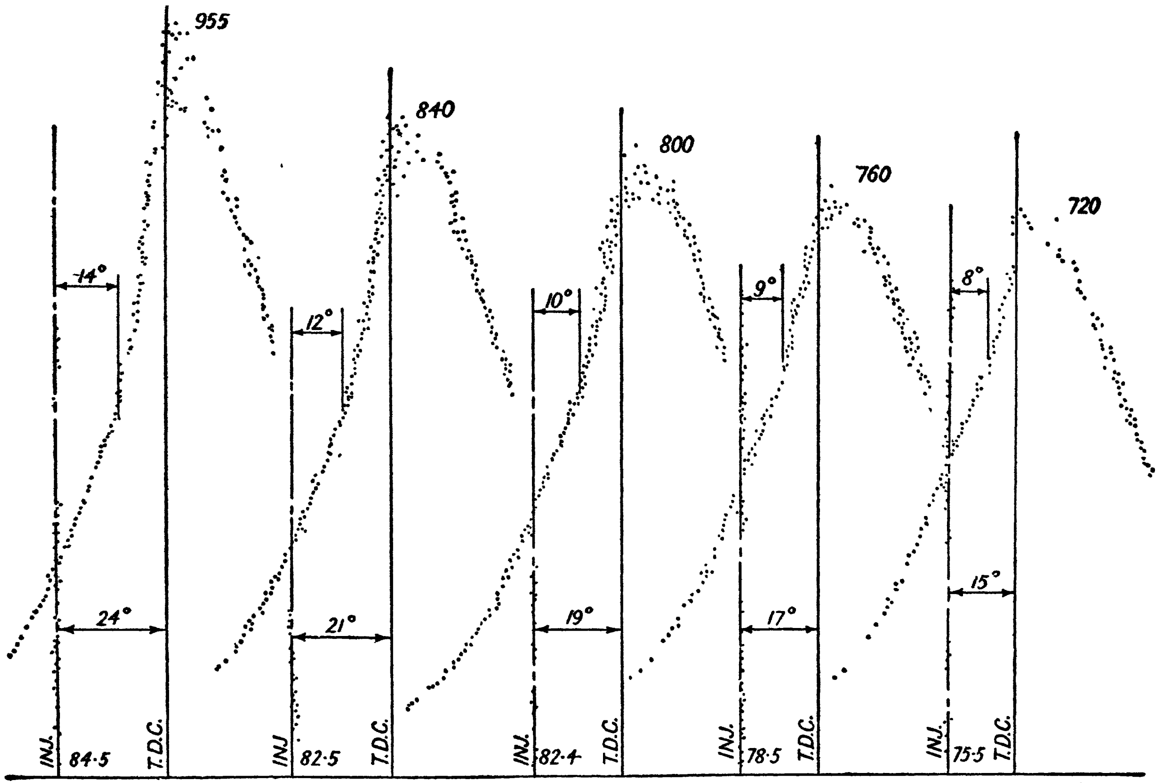
The usual range of the injection timing is from about 8° to 15° before T.D.C., with an extreme range of from about 5° to 20° or even a trifle earlier. At any given ratio this represents a change in density of the order of 50 per cent, and in temperature of the order of 20 per cent for the extreme range, and about half these amounts in the case of the more normal range of timing. These figures indicate the wide variation in conditions which can be introduced by a comparatively small change in injection timing. It must not be forgotten, however, that the effect of a given change in both temperature and density decreases in extent as the initial value is increased, so that the influence of a change in timing will be less marked with a high compression ratio than with a low.

The influence upon the delay period of a change in injection timing is well illustrated * by figs. 51, 52 and 53. These show the actual delay periods from three different combustion chambers, and are obtained from indicator diagrams taken by a Farnboro' indicator. All three sets of diagrams were taken at the same speed, and in the experiments no change was made apart from the alteration in timing and such adjustment to the dynamometer as was necessary to maintain the speed at a constant figure.

The results shown in fig. 51 are from an air cell combustion chamber which called for a considerable amount of injection advance in order to obtain the optimum results. In this particular instance the delay shows a steady increase as the injection is advanced, there being a delay of 8° with an advance of 15° and a delay of 14° with an advance of 24°. At the same time both the maximum pressure and the rate of pressure rise increase as the injection is advanced. The compression ratio, based on the total swept volume, was 15·5 : 1.

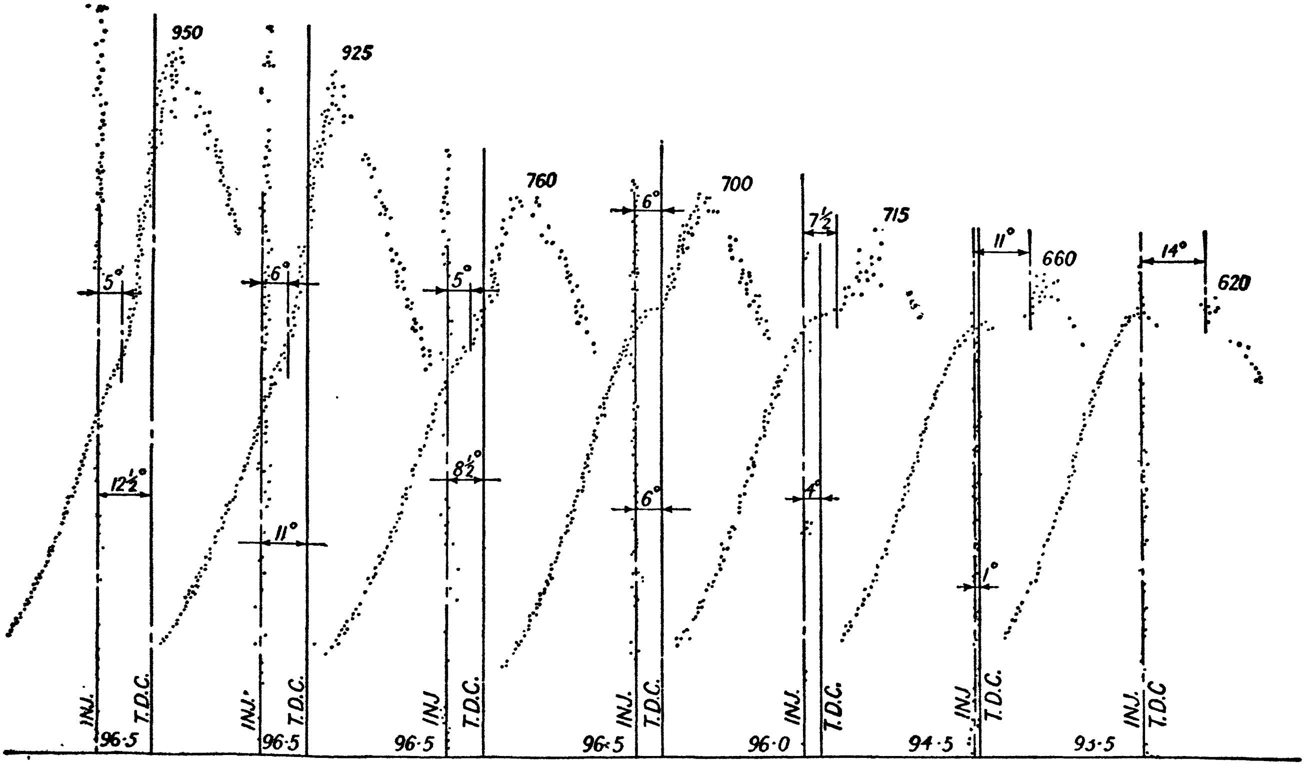
A directly opposite trend is shown in fig. 52, which gives the results obtained from a swirl chamber engine. Here the delay decreases from 14° when injection is timed to take place at T.D.C., falls to a value of 6° when the injection takes place at 6° before T.D.C., and then remains almost constant up to 12½° before T.D.C., which was the limit covered by the experiment. In this instance the rate of pressure rise and the maximum pressure reached increase as the injection is advanced, but whereas in the former case the B.M.E.P. increased steadily as the maximum pressure increased, the B.M.E.P. in the present instance attains its maximum value with the smallest amount of advance to give the minimum delay, and then with only

* Reproduced from *Proc. Inst. Auto. Eng.* (March, 1932).



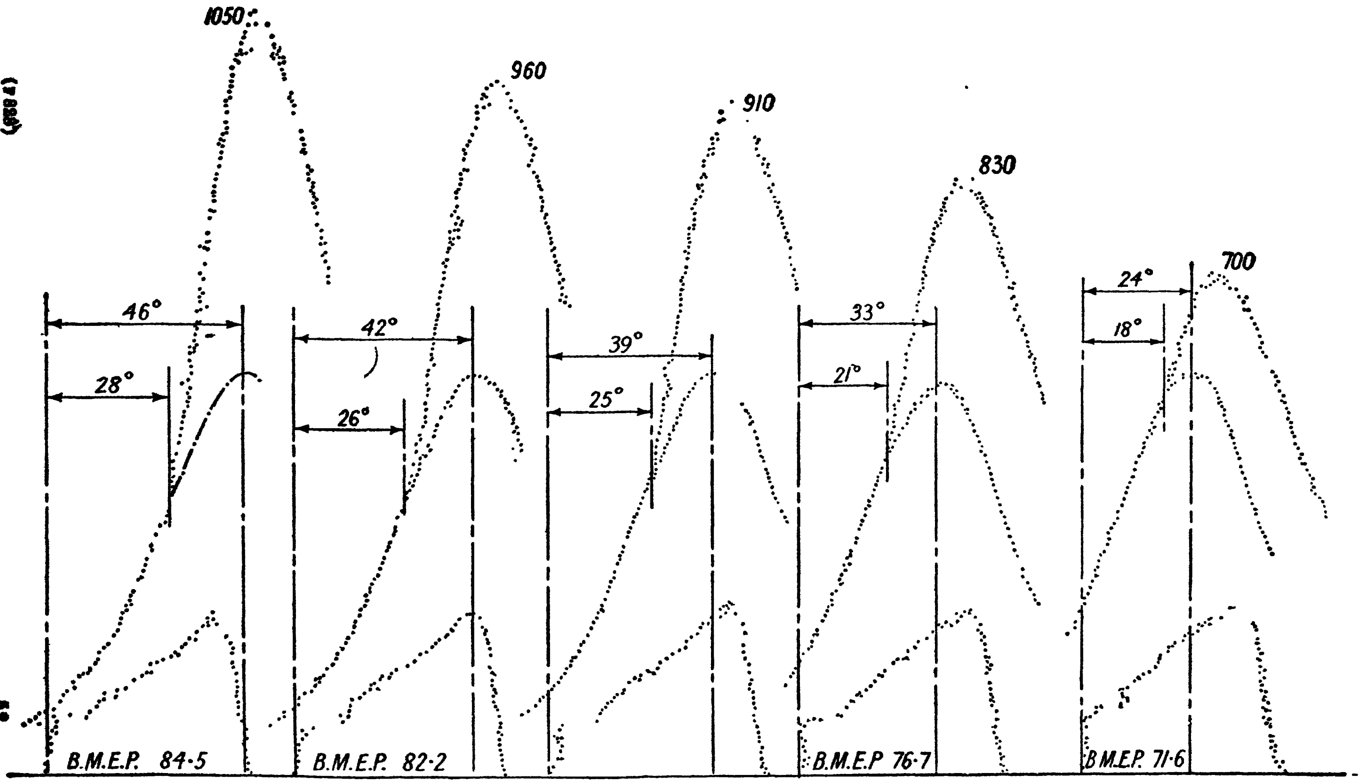
By courtesy of Inst. of Automobile Eng.

Fig. 51.—Variation in delay with a change in injection timing in an air cell combustion chamber



By courtesy of Inst. of Automobile Eng.

Fig. 52.—Variation in delay with a change in injection timing in a swirl chamber engine



By courtesy of Inst. of Automobile Eng.

Fig. 53.—Variation in delay with a change in injection timing in an open combustion chamber engine using an open injection nozzle

a moderate maximum pressure and rate of pressure rise, there being no need to adopt a high maximum pressure in order to obtain good economy. The tendency towards roughness, as indicated by a high rate of pressure rise, which accompanies an early injection timing is shown in both these figures. In the second example, however, an undesirably rapid rate of pressure rise is produced only with an injection timing which is a good deal earlier than the minimum advance necessary for maximum economy. The compression ratio in this instance was much the same as in the former case, being 16·2 : 1.

The diagrams in fig. 53 relate to an open type of combustion chamber which was fitted with a single-hole open type of fuel nozzle. Here again a very early injection is called for, the minimum advance in this series being equal to the maximum used in either of the preceding cases. Here the delay increases steadily as the injection is advanced, increasing from 18° for an injection timing of 24° early, to 28° with a timing of 46° early. The diagrams in this instance show the pressure on the fuel during injection and serve to give an indication of the much greater amount of fuel which is in the cylinder with a long delay than when the delay is reduced. Both the maximum pressure and the B.M.E.P. increase right up to the maximum advance used, but there is very little change in the rate of pressure rise, and although the maximum pressures recorded are decidedly high this engine was exceptionally smooth at all times. The economy, however, was decidedly poor, and although this is perhaps not the place to discuss this drawback, it may be mentioned that it was due to the coarse spray given by the open nozzle during the initial phase of injection. This condition results in a long delay but a slow rate of burning when ignition does take place. The compression ratio of this engine was 16 : 1.

The apparently contradictory nature of the results obtained from these three experiments is quite simply explained by the tremendous difference in the amount of injection advance required by each of the three engines. The advance required by the air cell engine is greatly in excess of that required by the swirl chamber engine, while that required by the open chamber is even greater than that required by the air cell engine. For the swirl chamber engine the minimum injection angle used was at T.D.C., and the temperature and density conditions were therefore deteriorating during the delay period, with the result that this was prolonged. As the injection was advanced the conditions became more favourable and the delay decreased progressively till a minimum figure was reached and the experiment, which in all three cases was conducted solely for the purpose of ascertaining the optimum injection setting, was terminated. Had the injection been advanced yet further the delay period would eventually have increased again.

As all three sets of figures were obtained from engines of substantially equal cylinder volumes running at the same speeds and having very nearly the same compression ratio, the results are more or less comparable, as is shown in fig. 54, where all three sets of figures

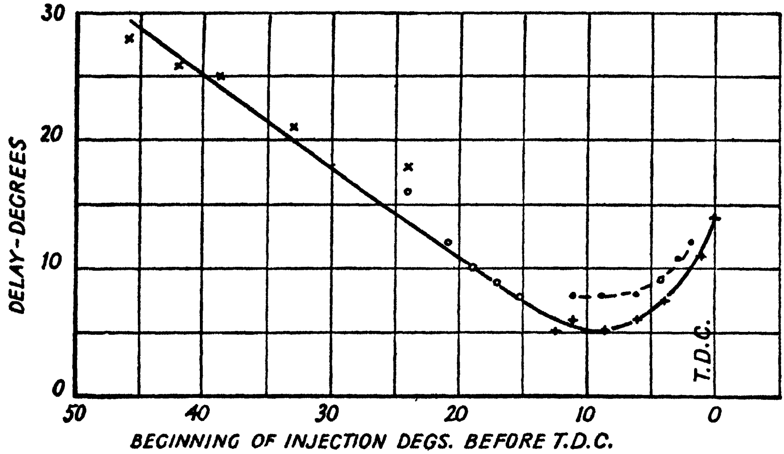


Fig. 54.—Variation in delay with injection advance

are plotted on a common basis and very nearly lie on a fair curve. This serves to illustrate the general relationship between injection advance and delay period. The fact that the three sets of readings lie upon a fair curve is perhaps rather a coincidence; some break in the continuity might reasonably have been expected, as shown by the short dotted curve which was obtained from a second swirl chamber engine having a slightly smaller cylinder capacity. But if the experiment had been tried of varying the injection timing of any one of the three engines over the total range covered by all three, the form of the delay curve would have been substantially that shown in fig. 54.

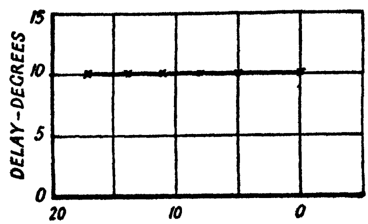


Fig. 55.—Delay and injection advance with an open chamber

Fig. 55 shows the results obtained from an open combustion chamber provided with a four-hole sprayer. This engine, which again had a compression ratio of 16 : 1, has a constant delay angle of 10° throughout the range covered by the experiment, i.e. from top dead centre to 16° before T.D.C. In this particular instance the fine spray which results from a high injection pressure—the nozzle opening pressure was set to 175 atmospheres—and a high rate of injection

undoubtedly combined with a high compression ratio to produce this lack of variation in delay period over the range covered by the experiment.

From the foregoing it will be clear that it is dangerous to generalize too much. Individual peculiarities of some engines in one direction may be sufficient to override some opposite tendency. It may, however, be said that fundamentally an increase in compression ratio at the moment at which injection commences, whether obtained by a change in the actual compression ratio of the engine, or by a change in injection timing, will tend to reduce the delay period, while a change in the opposite direction will have the reverse effect. The extent of the improvement from a given increase in compression ratio diminishes as the original ratio gets greater.

13. The Influence of Load upon Delay.

It is a fairly general characteristic of the high-speed compression-ignition engine that the smoothness of operation tends to improve as the load is increased. Further, an engine after being started up from cold will improve in smoothness during the first few minutes of running. This is due to the influence of temperature, and is usually most pronounced in engines which have a comparatively low compression ratio or in which the heat loss during compression is relatively high. Instances can be cited where a reduction in load has a smoothing effect, but this in no way affects the truth of the general statement just made. Engines which have this opposite tendency and decrease in smoothness with an increase in load will be found to have an abnormally long delay period. This means that over a wide range of loads, possibly even for the whole range from full load to no load, the whole of the fuel has been introduced before ignition takes place. As the load increases, therefore, the amount of fuel present at the moment of ignition, and therefore the amount of uncontrolled burning, increases and the tendency towards roughness increases also. Any reduction in delay which may result from the increased temperatures prevailing at the higher loads can have no influence towards improving the smoothness until the load has been increased to a point where ignition takes place before the whole charge of fuel has entered the cylinder. The use of a long delay fuel in an engine not prone to roughness may result in reversing its tendency, i.e. cause it to increase in roughness as the load increases instead of the opposite effect.

With a really long delay the smoothing effect of an increase in temperature may be so small as hardly to produce a noticeable result. Although, generally speaking, the roughness increases with the quantity of fuel delivered during the delay period, there is a point beyond which a further increase in delivery does not seem to produce a further increase in roughness. The influence of an increase in temperature

therefore will not make itself felt unless the quantity of fuel is less than this critical value.

In all high-speed engines there is a certain minimum load which must be carried before the injection period exceeds the delay period. Until this point is reached the tendency is for the amount of uncontrolled burning, and therefore the roughness, to increase. The value of this minimum load will, of course, depend upon the design of the individual engine and the fuel it is called upon to use. This point is well illustrated in fig. 56, which shows the nozzle valve lift diagram at different loads for an engine running at 1400 r.p.m. The average delivery per cycle and the delay angle as well as the load are shown

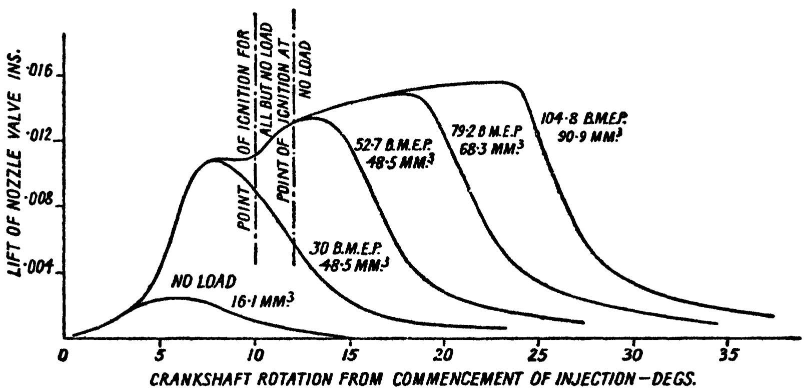


Fig. 56.—Showing how the delay period exceeds the injection period at light loads

for each diagram, and indicate that in this engine a quite considerable quantity of fuel is delivered during the delay period and that at all loads below about 30 lb./sq. in. B.M.E.P. the whole of the fuel has been received before ignition takes place. In this particular case, however, provided that a good jacket water temperature was maintained, the operation was by no means rough.

The variation in compression temperature under differing conditions of load may be very considerable. The compression temperatures in Table XII (p. 92) are based upon an assumed initial compression temperature of 340° abs. This figure allows for an average value of atmospheric temperature and a normal amount of heating during induction, and also for the heat received from the residual exhaust gases. Under cold starting conditions there will be no preheating of the air and no residual exhaust gases until the engine has actually been started, and at the same time the cold cylinder walls will mean a much greater heat loss during compression than the normal, resulting in a lower value of n during compression. When starting from cold, therefore, compression starts from atmospheric

temperature (which may be well below freezing-point), and this, coupled with the lower value of n, means that the final compression temperature will be much below that normally reached under average running conditions; less, in fact, than is reached even when running under no-load conditions.

A rough idea of the reduction in compression temperatures may be gained by comparing the two temperatures from which compression may start. Thus, if compression starts from a temperature of 0° C., 273° abs. instead of 340° abs. as was assumed for fig. 50, then the final absolute temperature will, at the least, be reduced in the ratio of the two initial temperatures, i.e. $273/340 = \cdot 803$, or to some 20 per cent less than that reached when running normally. This 20 per cent reduction is apart from any reduction that will be occasioned by any decrease in n.

This loss of compression temperature under cold starting conditions has a significance far greater than any momentary roughness in running which it may occasion. For any given engine using a particular fuel there will be a definite atmospheric temperature below which it will not be possible to start the engine from a cold state. unless it is possible to keep the engine turning over long enough to enable it to warm itself up by its own heat loss, or unless some extraneous source of heat is available.

With a reduction in compression temperature of the order indicated above, it is not surprising that immediately after being started from cold an engine should be noticeably less smooth than after it has been running for a few minutes. The effect upon the delay period is similar to that produced by Le Mesurier and Stansfield's experiment referred to on p. 113.

The conditions which exist under light-load running differ only in degree from those just outlined. It is true that a change in compression temperature of the magnitude of that just quoted can hardly result from a change in load, but in many instances the difference is amply sufficient to make a noticeable difference in the smoothness of the engine. For a given initial pressure an increase in compression temperature will, of course, be reflected in an increase in compression pressure, the increase in the absolute pressure being proportional to the increase in absolute temperature, and it is from the changes in pressure that the changes in temperature can be assessed once changes in volumetric efficiency and the temperature of the residual exhaust gases have been taken into account.

The changes in delay period with load and the corresponding changes in compression pressure and temperature are well shown by the figures in Tables XVI and XVII (pp. 131, 133). In the first of these, a fall in compression pressure of 40 lb./sq. in. was found to occur between full load and no load, while the delay increased from 8° to

12°, an increase of 50 per cent, the calculated difference in compression temperature being 126° C. In the second series no difference in compression pressure was recorded, but the calculated compression temperature changed nearly 80° C. between full load and no load, and the delay period increased from 9·5° to 11·5°, a little over 20 per cent.

In order to arrive at the compression temperatures the temperature at the commencement of compression was calculated from the volumetric efficiency of the engine at the load in question and the exhaust temperature at the same load, both of which were known, with the assumption that the exhaust gases were at atmospheric pressure at the beginning of the suction stroke. The equation $P_1V_1/T_1 = P_2V_2/T_2$ was then used for determining T_2. It is argued that these temperatures, although perhaps not absolutely accurate, have at least the correct relative values and serve to show very clearly the differences in compression temperature which can exist under different conditions of load.

Both engines had a compression ratio of 16 : 1, and the cylinder capacity was very nearly the same. The same fuel was used for each engine, and the speed in both instances was 1500 r.p.m., but different types of combustion chamber were used. The engine from which the figures in Table XVI were obtained was fitted with a high swirl chamber and was provided with an uncooled surface from which heat could be extracted by the air during compression. The engine for Table XVII had an open type of combustion chamber and was provided with only a moderate swirl.

TABLE XVI

Swirl Chamber Engine

B.M.E.P. lb./sq. in.	R.P.M. Actual	Delay Period		Comp. Press. lb./sq. in.	Temp. deg. abs.	
		deg.	sec.		Suction	Compression
95·2	1504	8	·000887	580	332	912
88·4	1507	8	·000885	578	329	899
81·2	1499	8·5	·000945	578	327	891
69·2	1505	10	·00111	560	323	855
55·8	1505	11	·00122	550	319	830
41·8	1507	11	·00122	550	316	821
21·7	1500	12	·00133	555	312	818
11·5	1522	12	·00131	550	310	805
0·0	1496	12	·00134	540	308	786

That the swirl chamber engine was successful in extracting a considerable amount of heat is clearly shown by the very much greater increase in compression temperature and pressure between no load and full load in this engine than that obtained with the open chamber.

At very small loads, however, the heat loss appears to be somewhat greater with the swirl chamber, as is shown by the lower temperatures and pressures a· the lower loads. The results from these two sets of

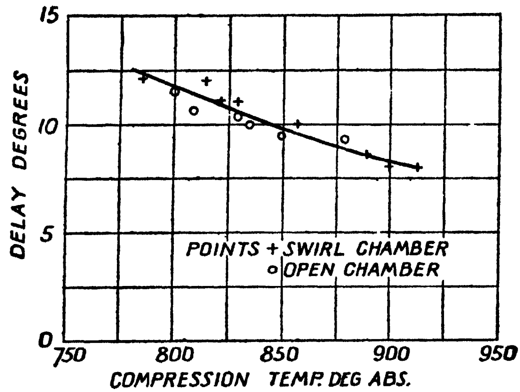


Fig. 57.—Influence of compression temperature on delay

figures are exhibited in fig. 57, which shows the delay angle plotted against compression temperature, and fig. 58, which shows both the delay angle and the compression temperature plotted against B.M.E.P.

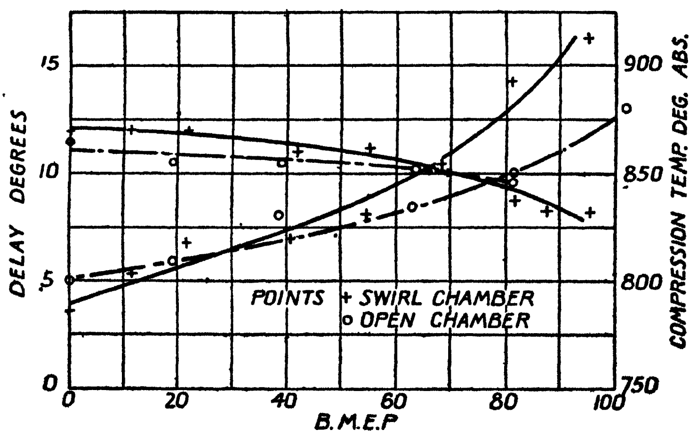


Fig. 58.—Influence of load on delay and compression temperature

If due allowance is made for the difficulty of measuring the delay period accurately and for the other factors incidental to such an experiment, there is a very good degree of agreement between the two sets of figures, and the figures clearly illustrate the influence upon

the delay arising from the differences in compression temperatures which can be produced by changes in load.

TABLE XVII

Open Chamber Engine

B.M.E.P. lb./sq. in	R.P.M. Actual	Delay Period		Comp. Press. lb./sq. in.	Temp. deg. abs.	
		deg.	sec.		Suction	Compression
102·3	1493	9·5	·00106	550	338	879
80·4	1494	9·5	·00106	550	326	848
63·1	1498	10·0	·00111	550	321	836
38·5	1507	10·5	·00116	550	316	831
19·2	1501	10·5	·00117	550	312	810
0·0	1501	11·5	·00127	550	308	800

The extent of the change in compression temperature and pressure will vary considerably with different designs, the compression ratio, the nature of the air movement and the presence of hot spots all exerting an influence, so that it is not possible to predict the extent of the change for any particular design.

The influence of this change in temperature upon the actual extent of the delay and the consequent effect upon the engine are difficult to predict with any exactitude, but generally speaking the amount of uncontrolled burning will increase with a decrease in load, and the tendency is therefore for a loss of smoothness at the lighter loads. Numerous proposals have been put forward for counteracting this effect. These have generally been associated with attempts to utilize heat from the exhaust gases to raise the initial temperature of compression. Unfortunately, however, under the conditions which make the use of some such device most desirable, i.e. lighter loads, the temperature of the exhaust gases is so low that the heat available is too small to produce any appreciable increase in compression temperature, so that such devices are ineffective. For engines which need some improvement an increase in compression ratio will usually be found much more efficacious.

14. Delay and Engine Speed.

The relationship between delay and engine speed is one of first importance if really high engine speeds are required, and assumes special significance for an engine which is required to deliver its full torque over a wide range of speeds.

To ensure satisfactory operation throughout the required speed range the cycle of operation must be reproduced with considerable exactitude at each and every speed at which the engine is called upon

to operate, and the all-round performance of an engine is intimately bound up with the success attained in this direction.

If a certain fuel when used in a given engine has a delay period of t sec. at n r.p.m., it follows that, unless some change in conditions takes place at the same time as a change in engine speed the delay will still occupy t sec. at $2n$ r.p.m. or at $n/2$ r.p.m. In other words, the delay period measured in degrees of crankshaft rotation will increase directly with the speed of rotation. The cycle of operation therefore will not be reproduced with any degree of exactitude at widely different speeds, and the engine behaviour and performance will undergo a material change as the speed changes.

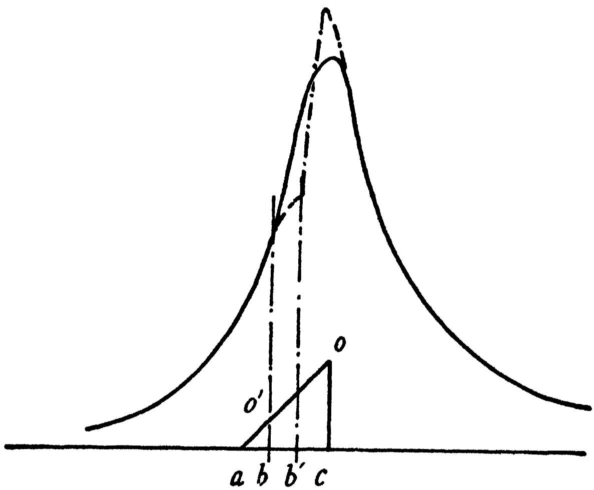


Fig. 59

Theoretically, for the cycle to be reproduced exactly at all speeds over a wide speed range the delay period when measured in degrees of crankshaft rotation should remain constant, i.e. when measured in time it must vary inversely as the engine speed. That this is so will be seen from the pressure-time diagram shown in fig. 59.

In this diagram, *a* represents the beginning of injection, *b* the point of first pressure rise above the compression pressure, and *c* the termination of injection. For purposes of illustration the rate of fuel delivery per degree of rotation has been assumed to be constant; the fuel delivered may therefore be represented by a straight line drawn from *a* to a point *o* above *c*, such that *co* represents to scale the total fuel charge. At *b*, the end of the delay period, a quantity of fuel equal to *bo′* will have been received, and, all else being equal, this quantity must be maintained constant regardless of speed if the pressure

is to remain unaltered. If, however, the delay period *ab* is increased to *ab′* owing to a change in speed, a greater quantity of fuel will have been received before ignition takes place, with the result that a greater rate of pressure rise and a higher maximum pressure will be produced.

From the theoretical standpoint there are no changes occurring with an alteration in speed which can influence the delay period, either in one way or the other. The engine measures out the same weight of air per cycle and subjects it to the same change in volume; the temperature and pressure at the end of compression thus remain unchanged and are independent of engine speed. Under practical conditions, however, there are certain changes in condition as the engine speed varies which have a direct influence upon the delay period. The compression is not truly adiabatic, some heat being lost, but as the speed increases the time taken for the compression stroke decreases, and there is therefore less time for heat to be lost, with the result that the compression temperature increases. Further, for the same weight of fuel burned per cycle the temperature of the cylinder and combustion chamber walls increases with the speed, and thus reduces still further the loss of heat during compression; at the same time this may result in a larger amount of heat being picked up by the air during the early part of compression, when the air temperature is less than that of the surrounding metal. Again, for a given fuel delivery, the temperature of the exhaust gases usually increases with the speed, so that the final temperature of the mixture of residual exhaust gases and the fresh air charge will be increased. On the other hand, it usually happens that the shorter time occupied by the induction stroke results in less heat being picked up by the air during admission. With the high compression ratios used for compression-ignition engines the volume of the residual exhaust gases is small, so that the increase in final suction temperature obtained from the higher exhaust gas temperature is small. This, however, is offset by the relatively high value of $r^{\gamma-1}$, and for each degree increase in final suction temperature the compression temperature will be increased by from 2·5° to 3° C.

By so arranging the combustion chamber that the air is brought during the compression stroke into close contact with the more highly heated parts of the combustion chamber, a very material increase in compression temperature can be produced. In certain chambers an uncooled portion is provided specially for this purpose, but even without this special provision quite a large increase in compression temperature can be obtained. In almost every engine there will almost certainly be some increase in compression temperature as the speed increases, but the extent of the increase will vary greatly with different designs. This increase in compression temperature is highly

desirable, and is particularly effective in keeping down the delay period as the speed increases, and it is probably not an exaggeration to say that without this it would be quite impossible to obtain really satisfactory operation over a wide range of speeds.

In addition to the influence of the increase in compression temperature, certain changes take place in the fuel delivery which, while they have no influence upon the duration of the delay period, are of material assistance in offsetting the adverse effects of an increase in the delay period. The compressibility of the fuel, and a certain lack of rigidity on the part of the pump mechanism and its driving gear, produce a material difference between the actual duration of the injection period and that given by the points of opening and closing of the ports, or valves, of the fuel pump. This departure of the injection period from its theoretical duration has not a constant value but changes with speed, the actual injection period when measured in degrees of crankshaft rotation increasing as the speed increases. The fuel delivered during the delay period does not, therefore, increase in direct proportion to the duration of the delay period, but at some lower rate, with the result that the amount of uncontrolled burning which takes place is not increased in proportion to the duration of the delay period.

The advantages of this will be obvious, and under practical conditions this prolongation of the injection period is sufficiently great to have a marked influence upon the actual conditions prevailing during the initial stages of combustion. This is particularly so in cases where high injection pressures are employed, and where nozzles are used which offer a high resistance to the flow of the fuel. This is one of the factors which have to be taken into account during the development of an engine, because it will be obvious that, no matter how advantageous its effects may be during the delay period, if the increase in duration of injection is too prolonged at the higher speeds it may become disadvantageous to the efficiency of the cycle by causing late burning of the fuel and hence a reduction in the effective expansion ratio.

An instance of a marked increase in compression temperature with an increase in speed is given in Table XVIII. These results were obtained from a single-cylinder experimental engine fitted with a Ricardo Comet Mark I combustion chamber having a compression ratio of 16 : 1.

The compression pressures were measured inside the swirl chamber and not immediately above the piston, so that there can be no possibility of the increase in pressures having been caused by the pressure required to force the air through the orifice and into the swirl chamber.

It will be observed that the compression pressure increased from 507 lb./sq. in. G. at 500 r.p.m. to 622 lb./sq. in. G. at 2000 r.p.m.,

an increase in pressure of 22 per cent on the absolute scale with, approximately, the same increase in the absolute compression temperature.

The data for making a calculation of the actual compression temperatures at the various speeds are not available, but from the evidence available the change in volumetric efficiency is small, being reduced only slightly at the higher speeds. It would appear also that the initial compression temperature will not vary greatly; the initial air temperature tends to fall slightly with speed (see p. 135), while the temperature of the residual exhaust gases will increase with speed, and it may reasonably be assumed that in this instance the change in absolute compression temperature is of the same order as the change in absolute compression pressure. Figures calculated on this basis are included in Table XVIII.

TABLE XVIII

R.P.M.	Comp. Press. lb./sq. in. G.	Delay		Approx. Comp. Temp. deg. abs.
		deg.	sec.	
493	507	6	·00203	892
747	515	6	·00134	920
991	545	6	·001009	940
1294	565	7	·0009	990
1491	575	7	·00078	1010
1790	610	8	·000745	1070
1992	622	8·5	·000711	1090

Measured in degrees the delay has increased from 6° to 8½°, but in time it has decreased from ·00203 sec. to ·000711 sec., a reduction of 65 per cent. Expressed as a percentage, the delay in degrees actually increased by 42 per cent; this sounds large, but an increase in delay of only 2½° for an increase in speed of from 500 r.p.m. to 2000 r.p.m. cannot be considered as very great, nor can its effects be very serious.

The figures given in Table XVIII are plotted in fig. 60 (p. 138), and indicate that over the range of speeds covered by the experiment the compression pressure increases directly with the speed. The delay in degrees does not give too good a curve; this is due to the difficulty of measuring the delay really accurately, the difficulty being increased by a slight shift in the actual injection point at certain speeds. The true tendency is probably from a nearly constant value at the low speeds to an increasing upward trend as the speed increases. Measured in terms of time, the delay curve falls rapidly at first, but later flattens out and indicates a probable minimum value of about ·0007 sec.

These results go to show that this type of combustion chamber has a very marked influence upon the compression temperature, and

that the air must have extracted a very appreciable quantity of heat during compression and thereby obtains over a wide range of speeds a very fair measure of compensation for the effect of speed upon the delay period. From the trend of the curve it would appear that a delay period of about 12° might be expected at a speed of 3000 r.p.m.,

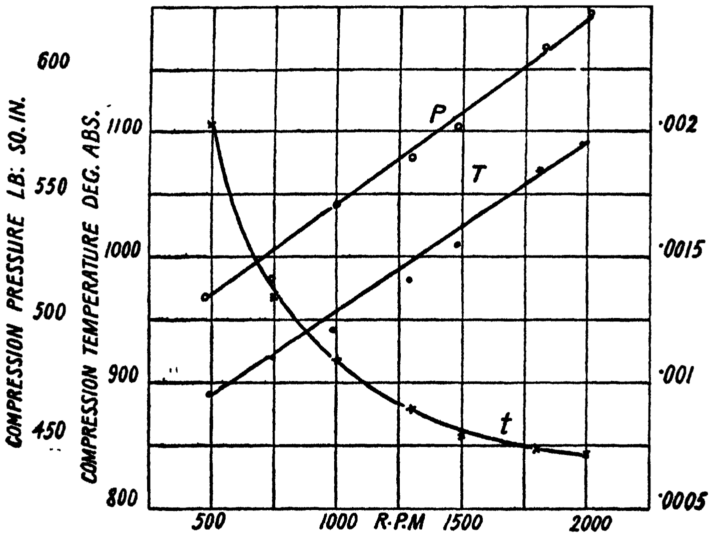


Fig. 60.—Variation of compression pressure and temperature and of delay period with speed in a swirl chamber engine

a figure which is still within the region of satisfactory operation, as is shown by the fact that many engines give quite satisfactory operation with a delay period of this order at little more than half the above speed.

Figures of a similar character but obtained from an engine slightly larger than in the previous case and fitted with an open type of combustion chamber are given in Table XIX.

TABLE XIX

R.P.M.	B.M.E.P. lb./sq. in. G.	Comp. Press. lb./sq. in. G.	Delay		Duration of Injection deg.	Comp. Temp. deg. abs.
			deg.	sec.		
517	94	525	5	·00161	12	900
765	98·6	525	6	·00131	14	900
1003	98·8	525	7½	·00121	14	894
1250	97·1	550	8½	·00114	18	941
1500	97·3	570	10	·00111	20	976
1770	92·7	570	11	·00104	22	1010
1955	88·7	595	12	·00102	24	1070

This engine has a much smaller increase in compression pressure as the speed increases, the increase being only 70 lb./sq. in. between 500 and 2000 r.p.m. against 115 lb./sq. in. in the previous case. The delay increases from 5° at 500 r.p.m. to 12° at 2000, but measured in time, decreases from ·00161 sec. to ·00106 sec. On the time basis the delay shows the same tendency to decrease to a minimum value; the minimum in this engine, however, appears to be about ·001 sec., instead of ·0007 sec. in the previous case. The figures from Table XIX are shown plotted in fig. 61, which gives the delay angle, delay time, and compression pressure and temperature plotted against speed.

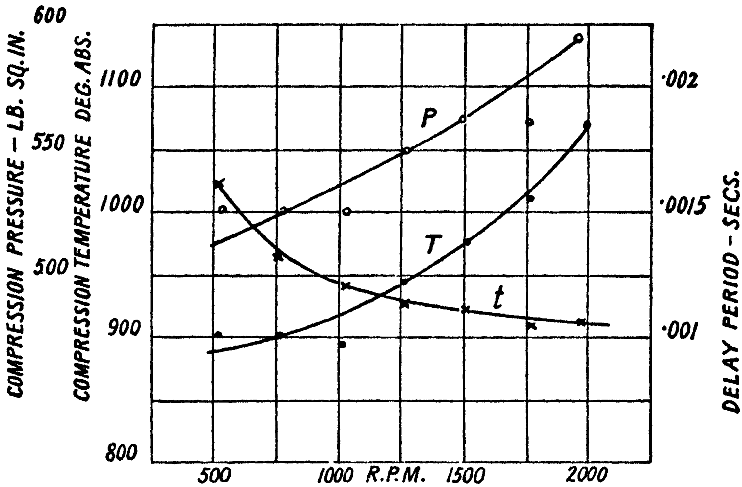


Fig. 61.—Variation of compression pressure and temperature and of delay period with speed in an open combustion chamber engine

For this engine data for the volumetric efficiency and exhaust temperatures were available, and it is therefore possible to arrive at figures for the compression temperatures which can be considered as being fairly close to the actual values. The figures so obtained have been included in both Table XIX and fig. 61. These, with the exception of two readings at the low speed end, produce a fair curve and indicate a steady increase in temperature as the speed increases. How far this trend will continue cannot be said, but obviously there must be a limit to this increase in temperature. The delay, measured in time, is shown plotted against the compression temperature in fig. 62. In plotting this latter curve the compression temperatures as read from the curve have been used for the two points which do not lie on the curve. From this it will be seen that there is a steady decrease in the effectiveness of a given increase in temperature, precisely as was indicated by Bird's experiments referred to on p. 111. This goes

to prove that the tendency indicated by bomb experiments is correct, even if the actual values obtained in the bomb bear little relation to those found in the engine. It may be mentioned that the curve in fig. 62 shows the effect of a variation in temperature alone; the change in volumetric efficiency over the range of speeds covered by the experiment was so small that the density may be considered as having remained constant at all speeds.

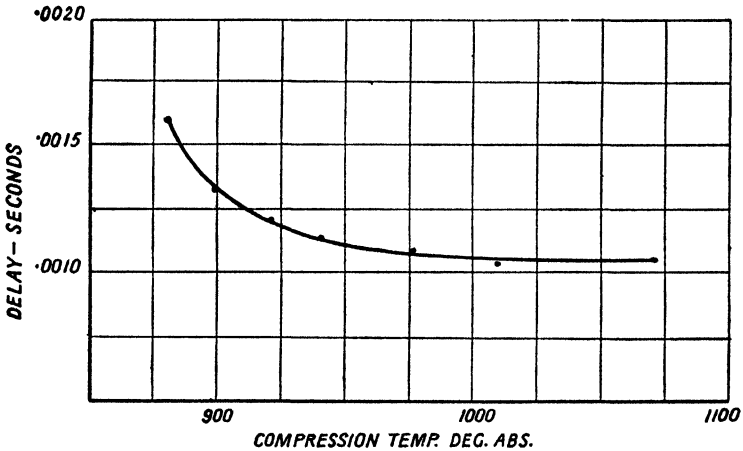


Fig. 62.—Variation in delay with compression temperature in an open chamber

This set of figures serves also to illustrate the change in the length of the injection period as the speed increases and the consequent effect upon the fuel received during the delay period. The duration of injection was measured from "rate of change" diagrams taken from the movement of the spray valve, and these readings are given in Table XX, together with the actual quantity of fuel delivered per injection, the mean delivery per degree, and also the apparent delivery during the delay period obtained from the delay and the mean delivery.

TABLE XX

R.P.M.	Fuel per Injection, m/m³	Duration of Injection, deg.	Mean Rate of Injection, m/m³/deg.	Apparent Delivery during Delay, m/m³
517	79·0	12	6·58	32·9
765	82·8	14	5·9	35·4
1003	82·1	14 (?16)	5·9 (?5·13)	42·8 (?37·2)
1250	81·3	18	4·5	38·3
1500	81·8	20	4·09	40·9
1770	79·6	22	3·62	39·8
1955	79·0	24	3·29	39·5

The mean rate per degree and the apparent delivery during the delay must not be taken as being absolute values. The delivery is not at a uniform rate during the injection period but increases as the injection proceeds. This is clearly indicated by the form of the injection pressure diagram, which shows a steadily increasing pressure right up to the point at which the sudden collapse takes place. The figures, however, do serve to show the general tendency and indicate that the delivery during the delay will not increase in proportion to the increase in the delay period.

In this particular instance the increase in delay period was roughly proportional to the increase in injection period, so that it is reasonable to assume that the quantity of fuel delivered during the delay was roughly the same at all speeds, a contention which is supported by the apparent delivery figures given in the table. This comparatively small change in the quantity of fuel delivered during the delay is of very great assistance in preventing a large change in the conditions under which the fuel is burned. The fact that the duration of injection has increased from 12° to 24° between 500 and 2000 r.p.m. with a corresponding reduction in the rate of injection is of far less importance than a change in delay of from 5° to 12° over the same range of speeds would be if the injection period remained constant at 12°. In the latter event at a speed of 2000 r.p.m. the whole of the fuel would have been delivered before ignition occurred and the whole of the combustion would therefore have been uncontrolled.

The influence of temperature upon delay time under actual engine conditions is shown in fig. 63. This incorporates all the information already given, both for a constant speed with varying torque and a constant torque with varying speed. Le Mesurier and Stansfield's figures for differing fuels are given also.

In each instance the temperature is the chief variable, the density remaining approximately constant in each individual case, although in the case of Le Mesurier and Stansfield's figures a moderate change in density will have been produced by their method of changing the temperature. The temperatures shown are those calculated for the end of compression, because information was not available to enable the temperature to be determined other than at the end of compression. It is of course the mean of the temperature between the commencement of injection and the moment of ignition that is the real criterion, but this will be only slightly less than that at the end of compression, and the form of the curves will not be altered.

The curves show the same trend as was noted by Bird in his bomb experiments, namely, a tendency for the delay time to fall to a constant value when the temperature is above a certain figure. In some of the examples, the maximum temperature reached was a good way short of the figure at which the minimum delay would be reached, and

it is not possible to extrapolate the curve so as to be able to say with any degree of accuracy what the minimum time would be; it appears, however, that the minimum will not differ very greatly for any of the instances given.

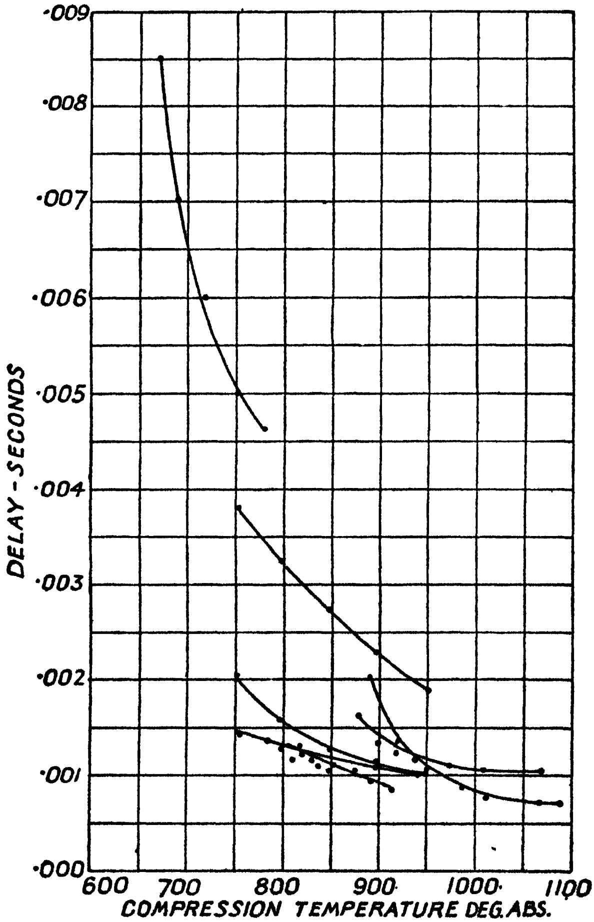


Fig. 63.—Relationship between delay period and compression temperature: note the similarity between results obtained under differing conditions

That the delay time can be less than ·001 sec. is clear from the evidence provided, and the general trend of the curves indicates a minimum value of ·0005 sec. or a trifle higher. It would, of course, be unwise to dogmatize as to the minimum value which can be attained under any set of circumstances. It will be clear that the delay can never be reduced to zero, because even if the fuel on entry were already

at its ignition temperature the fuel and air must attain a certain intimacy of contact before combustion can occur, and this will require a certain amount of time. The very definition of delay time, the interval which elapses between the moment at which the fuel enters the combustion chamber and that at which the first rise of pressure above the compression pressure takes place, shows the impossibility of reducing the value to zero. Sufficient heat must be released to make its presence felt, and the wave of pressure produced thereby has to travel through the mass of compressed gas and reach the recording instrument. The sensitiveness of the instrument and its location with respect to the point at which ignition originates are consequently factors in the determination of the delay time.

The results shown in fig. 63 were obtained from engines having differing compression ratios, but the trend of the curves suggests that approximately the same minimum delay time would be reached; a very much higher temperature, however, would be required with a lower compression ratio, which emphasizes the influence of an increase in density in reducing the delay time.

The figures suggest that in order that the delay time may be reduced to a value suitable for operating engines at high speeds the compression temperature should be not less than about 800° abs., say 500° C., and that it may, with advantage, be brought up to as high as 700° C. or even 800° C., especially if really high speeds are required. In fact, these latter temperatures would appear to be essential if a reasonable measure of combustion control is to be maintained at high engine revolutions.

It might be argued that the increase in compression ratio necessary to produce such temperatures will result in an unduly heavy engine and so limit the speed of rotation; on the contrary, however, the maximum pressures produced with a lower compression ratio and a longer delay are frequently greater than those obtained with a higher ratio and a reduced delay, while the shock which accompanies a large proportion of uncontrolled burning is far more detrimental to bearings, &c., than the somewhat greater average loading which accompanies a higher compression ratio.

The optimum compression ratio for an engine will be found to depend very largely upon the size of the individual cylinders. For cylinders of diameter in the neighbourhood of 4 to 5 in. a ratio of 15 or 16 : 1 would seem suitable in that the delay time is kept low and the maximum pressures can be kept within reasonable limits, i.e. below 1000 lb./sq. in., without serious sacrifices of efficiency. For smaller cylinders a ratio of 18 : 1, or even higher when quite small cylinders are used, may be necessary to offset the higher heat losses of the small cylinder, whereas with cylinders of larger size a somewhat lower ratio will probably be necessary in order to keep the

maximum loading down to more reasonable figures. The reduced heat loss with the larger cylinders, coupled with the fact that the maximum rotational speed is usually less with a larger cylinder, helps to offset any disadvantage which may result from the lower ratio. A ratio of about 12 appears to be the lower limit for the larger sizes of high-speed engines.

CHAPTER VII

Air Movement in the Compression-Ignition Engine

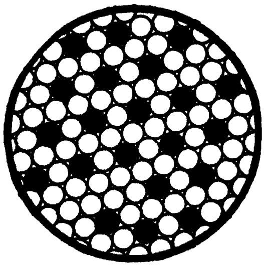
1. The Function of Air Movement.

In the account of the combustion of a single fuel particle the latter was likened to a miniature meteorite rushing across the combustion chamber, collecting oxygen, and being consumed by it. This meteorite must be consumed in a single journey from its source, the spray nozzle, to the far combustion chamber wall. If it strikes the wall it may be chilled and extinguished. If it wets the surface of the wall it will neither be in a state well adapted to securing oxygen expeditiously nor will the conditions in its neighbourhood be conducive to rapid and clean combustion because the layer of gas adjacent to the wall is subject to intense cooling and is therefore at a relatively low temperature, and also because this layer is comparatively stagnant. Under certain circumstances the fuel particle may strike the wall and rebound to return again on its journey across the chamber, but it is obviously better if matters can be so adjusted that this does not occur.

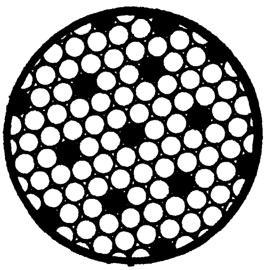
Although the fuel is in an exceedingly fine state of subdivision, a particle of it is obviously incomparably larger than a vapour molecule, and is therefore not in a state which will enable it to be satisfied by the first few molecules of oxygen with which it comes into contact. Even the smallest liquid particle is a relatively large mass, which will require a great number of molecules of oxygen before it can be completely satisfied. In order that our " meteorite " may be rapidly and completely consumed, it must be brought in contact with a steady stream of fresh oxygen molecules, just as a real meteorite is when moving through the earth's atmosphere. Whereas a real meteorite usually arrives alone, or at the most in very small numbers, and has unlimited supplies of oxygen upon which to draw, the fuel particle arrives in company with a dense crowd of others, numbering tens of thousands, in the combustion chamber, where only a strictly limited supply of oxygen is available. The actual purpose of this crowd of fuel particles is to use up, if possible, the whole of the limited supply of oxygen, or failing this, as large a proportion of it as it can do with efficiency.

Under optimum conditions the number of oxygen molecules met is, roughly, one out of every five, and the first fuel particles to arrive have a relatively easy task in finding sufficient for their requirements. As the process of combustion proceeds, however, the task becomes progressively more difficult, and the later arrivals, together with the remnants of the larger particles which have taken a longer time to burn, have an ever increasing difficulty in obtaining their oxygen.

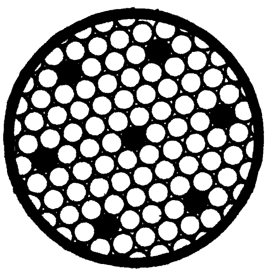
The rate of burning is governed entirely by the rate at which the



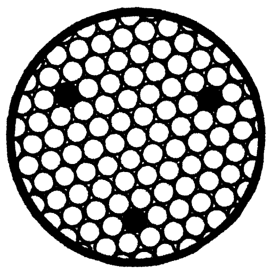
A. Pure air



B. 46% oxygen used



C. 70% oxygen used



D. 87% oxygen used

Fig. 64.—Illustrating the relative frequency of oxygen molecules with different degrees of air utilization

fuel and oxygen come into contact with one another, so that as more and more fuel is received the rate of burning gets progressively slower, until a point is reached when further supplies of fuel fail to find oxygen in time for the resultant heat to be used effectively. Ultimately a point is reached when further supplies of fuel are not burned at all, not necessarily because no more oxygen is present, but because it is so scarce that the fuel has been unable to find it in the short time available. The increasing difficulty experienced by the fuel in finding its oxygen supply is well illustrated by fig. 64, which is intended to represent a layer of air one molecule thick. Here the oxygen molecules are represented by the dark circles and the inert gases by the light circles. The relative frequency of the oxygen in pure air

is illustrated by diagram A, while B, C and D show the oxygen remaining after increasing proportions of the oxygen have been consumed. Diagram D represents the condition of the atmosphere in the cylinder when the oxygen content has been reduced to about the minimum found to be practical under present-day conditions of engine development. From these diagrams it is not difficult to realize the impossibility of making use of the whole of the oxygen, as well as the necessity for arranging matters so that the air and the fuel are brought together in some well-ordered manner if the best possible use is to be made of the oxygen.

To do this the whole of the combustion chamber volume must be quickly and thoroughly explored by the fuel. This thorough exploration offers certain difficulties; for practical reasons the fuel is admitted through a single nozzle (in a few instances more than one nozzle has been employed), situated at what is considered to be the most advantageous position. For the moment let us assume that the combustion chamber has been given what is probably the simplest and most straightforward shape—a hemisphere with the nozzle situated at the centre—and that the air in the hemisphere is in a quiescent condition. The problem is to distribute the fuel uniformly throughout this hemispherical mass of air, and to make the maximum possible use of the oxygen it contains. It is not only a matter of distributing the fuel in such a way that if no ignition took place and the whole of the fuel were to travel to the outer boundary, the fuel would then be uniformly distributed over the surface of the hemisphere (a matter which of itself would present considerable practical difficulties), but in such a way that each small volume of air receives and consumes its correct quota of fuel. It is a *volumetric* distribution of the fuel and not a *planar* distribution that is required. This distribution must take place during the actual process of combustion, since there can be no question of first producing a thorough mixture of fuel and air and then igniting it afterwards.

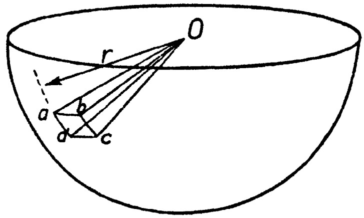


Fig 65

Suppose that a nozzle is available which will deliver the fuel in such a manner that the distribution is uniform when referred to the surface of the hemisphere, and consider what will take place in the wedge-shaped volume of air formed by a small square on the surface of the hemisphere and the radii joining the four corners of the square to the nozzle (fig. 65).

In accordance with our assumption the fuel is delivered in such a manner that it is uniformly distributed over the surface of the square

abcd at a distance *r* from the centre O. If no combustion takes place, the density of the fuel per unit area decreases in proportion to the square of the distance from O. Thus at half the distance from the centre the density of the fuel will be four times that at the outer surface of the hemisphere. As the fuel is in droplets of finite size which travel radially outwards from the centre, the distance between individual droplets will increase in direct proportion with the distance from the centre, so that as the distance from the nozzle increases, the possibility of the fuel particles failing to make contact with some of the available oxygen molecules increases also. This is even more apparent if the matter is considered on the basis of volumes. The volume of the wedge is a function of the cube of the radius, so that the volume of a wedge to a point half-way to the surface of the hemisphere is only one-eighth of that of the wedge which extends all the way to the wall of the hemisphere, while at the same time the whole of the fuel which is intended for the air beyond this half-way point has first to pass through this smaller pyramid of air. Thus, although it is fairly easy to search out thoroughly the zones close to the nozzle, the difficulty increases as the distance from the nozzle increases, both on account of the dispersal of the spray itself and because the volume of air to be explored increases with the cube of the distance from the nozzle.

Under operating conditions the fuel particles, or at any rate the first-comers, are progressively consumed as they travel towards the outer boundary and thus get smaller and smaller. They are by no means all the same size, and many of the smaller ones may reasonably be assumed to have been entirely consumed well before they could reach the outer surface. Since we are assuming more or less ideal conditions, let us assume that all the largest particles are completely consumed just as they reach the outer boundary. Under these conditions, not only do the particles get further and further apart as they travel outwards from the nozzle, but their separation is exaggerated by their decrease in size and number with the distance travelled. It is true that as the oxygen in the inner zone is used up the combustion of the particles is delayed until oxygen becomes available in the more remote regions, but since the fuel particles can only make contact with oxygen molecules which lie directly in their path, this will not alleviate matters to any very great extent.

It would thus appear that the form of chamber which we have been considering, i.e. a hemispherical chamber filled with quiescent air and provided with a fuel nozzle at the geometric centre, is exactly the opposite of what it should be. What is really wanted is a hemispherical combustion chamber into which the fuel is delivered radially *inwards* and uniformly from the whole of the outer boundary surface. We should then have ideal conditions. Our fuel particles, conditions being ideal, would all be of equal size and of such dimen-

sions that the oxygen molecules could not slip by between them; they would be consumed at equal rates and in such a manner that their mass would be a function of the cube of their distance from the centre and would reach zero just as the centre was reached. Under these circumstances we could even allow some difference in size of the fuel particles, the smaller ones being completely consumed after travelling only part of the way towards the centre, the larger ones still being completely consumed just as the centre is reached. With such an arrangement the problem of making effective use of the air would be simple but, as in the case of all other ideals, neither this, nor even a rough approximation to it, is attainable.

We have, therefore, to place the nozzle at the centre of the sphere and make shift with the arrangement first described, or rather what is only a very rough approximation to it, because nothing even approaching a nozzle which will give a hemispherical distribution, let alone a uniform hemispherical distribution, has yet been produced.

Under practical conditions the spray we have to employ is mainly conical in form, the angle of the cone being very much less than the 180° which would be necessary in order to fill a hemispherical chamber. The sprays may be one or more in number according to circumstances, but the maximum number employed is always comparatively low, especially in the case of the smaller engines. The number is usually limited by practical conditions, such as the necessity of maintaining a certain minimum pressure on the fuel at the nozzle and the difficulty of producing holes of less than a certain size. At best, something very much less than a hemispherical distribution is obtained, and naturally the difficulty of adequately exploring the whole volume of the combustion chamber is greatly accentuated.

Since we are apparently unable to make the fuel search out its own oxygen as thoroughly as is necessary or desirable, we have to resort to other means for doing that part of the task which the fuel fails to perform. If the fuel will not, or rather cannot, go after the air, then the obvious thing to do is to make the air go after the fuel. Air is by nature far more mobile than the fuel, and, moreover, can be readily induced to move in almost any desired path within reason. The fuel, on the other hand, can be given a definite movement in a straight line only. It is true that as the velocity of its arrival is dissipated it will deviate from the straight line and wander in a curved path, but the only positive and controlled movement which can be imparted to it is a straight-line movement. Further, apart from the possibility of splashing off the chamber walls, the fuel can be made to traverse the combustion chamber but once; the air, on the other hand, can be made to travel around the combustion chamber several times during the critical period if this is considered necessary.

2. Indiscriminate Turbulence.

In the foregoing discussion on the distribution of the fuel throughout the combustion chamber it was assumed that the air was in a quiescent state and the turbulence which inevitably follows the flow of a gas from one chamber to another was ignored. The very act of filling the cylinder with a fresh charge of air produces some degree of turbulence. This turbulence, however, is of an indiscriminate nature and is in the form of whirls and eddies, and although this form of air movement is admirably adapted to promoting rapid combustion in a carburetting engine, it is not that best suited to the needs of a compression-ignition engine.

Turbulence in a carburetting engine has a function fundamentally different from that required for a compression-ignition engine. The action of turbulence in a carburetting engine is to cause a thorough and continuous re-mixing of the gas and fuel vapour during the combustion period, with the result that the rate of inflammation is greatly increased owing to burning particles of mixture being hurled through regions which have not yet become ignited, helping to spread the flame and in this way bringing about rapid and complete combustion. The fuel and the air being already thoroughly mixed before ignition takes place, the turbulence is not required to perform the function of bringing the fuel and air into contact with one another, although it is of undoubted value in ensuring a thorough mixing of the two ingredients during the period preceding ignition. The turbulence being produced by the mixture flowing into the cylinder, it follows that the amount of turbulence, or rate of mixing of the charge, is proportional to the engine speed, and the rate of combustion will thus be increased automatically as the engine speed increases. Should the increase in turbulence not be sufficient at the higher speeds, a change in ignition timing provides a ready means for making up for any deficiency.

It will easily be understood that an air movement of an indiscriminate nature is not the one best suited to ensure that each burning fuel particle is met with a steady stream of fresh oxygen molecules until it is completely consumed. It is true that an eddying movement may help to disperse a zone of used-up air near the nozzle, but it is equally capable of removing a body of unused air from the path of the spray and substituting used-up air from another part of the combustion chamber, or of carrying a body of used-up air along with the spray. An indiscriminate air movement is therefore not suitable if a high output is required, although under certain circumstances it works quite well when only a moderate output is required from medium and slow-speed engines.

3. Air Swirl.

For high outputs at high speeds the air movement required is one that will remove the products of combustion from the neighbourhood of the fuel particles and continue to provide a supply of fresh air until combustion is completed. If this can be done, then the problem of rapid and complete combustion is greatly simplified. The fuel enters the combustion chamber as a stream of particles, with the natural result that the first-comers seize upon the oxygen and leave nothing but inert gases in the path of the later arrivals. The effective way to overcome the difficulty is to cause the air to move in a direction at right angles to that in which the fuel is travelling. To produce an air movement with or against the spray will result either in the fuel particles travelling along with their own products of combustion, or in the products from those ahead being carried back on to those behind, so that there will be little or no improvement over the conditions existing when no air movement at all is provided. What is wanted, therefore, is that the air shall move in a smooth and orderly stream at right angles to the direction taken by the fuel, as shown in fig. 66. Here AB represents the track of the fuel particles, which get smaller and smaller as they travel towards B; AC is the track of the air stream at right angles to that of the fuel. The products of combustion of each particle are carried off to leeward and out of the track of its successor, which thus meets with fresh pure air throughout the whole of its journey.

Fortunately, it is a simple matter to produce this movement within the engine. All that is necessary is to give the air a rotary motion within the combustion chamber and to place the nozzle so that the fuel is delivered either radially outwards from the centre of rotation, inwards towards the centre, or in a direction parallel to (but at a point some distance from) the axis of rotation.

Conditions are not, of course, quite so straightforward as is suggested by fig. 66. The fuel does not issue from the nozzle in a single row of globules, but in a jet which on leaving the orifice breaks into a conical stream of particles. There are, therefore, many particles which are to leeward of others and must therefore to some extent receive air contaminated with products of combustion. At the worst, however, the number of particles to windward of the most leeward particle will be infinitesimal compared with the number if no such air movement were provided. A further departure from fig. 66 is that the fuel particles do not maintain a straight path, but as they lose their velocity they are deviated and cause the spray to "bush" out at its head. This latter condition assists the spray to seek out the oxygen for itself, and is therefore beneficial when no definite air movement is provided; at the same time, it does not appear to have any adverse

effect when a definite air movement is provided, because a dispersal of the fuel in a direction at right angles to the direction of travel of both the air and spray assists the fuel in finding oxygen by reaching upwards and downwards into layers of air which otherwise might pass above or below, instead of through, the spray.

To enable this rotary movement of the air to be used to best advantage, the shape of the combustion chamber must be one well suited to the free movement of the air, that is, substantially circular in the plane in which the air rotates, a requirement which is easily fulfilled.

Being strictly limited in quantity, the air must have its movement accurately matched with that of the fuel. It will serve no useful purpose to give the air a rotary movement which bears no relationship

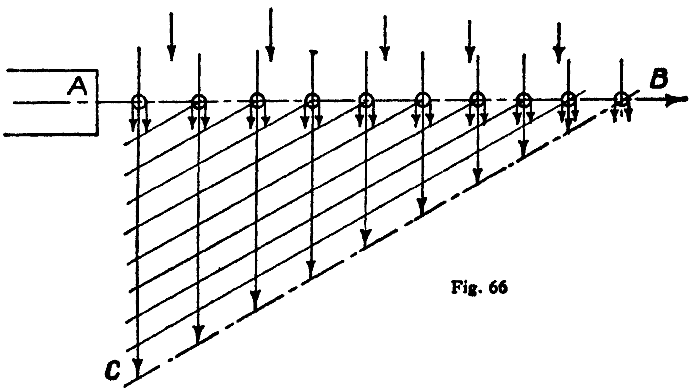


Fig. 66

to the speed at which the fuel issues from the nozzle. The number of sprays used is a governing factor in the correct rate of swirl, as this rotary movement is generally termed. This will be made clear from fig. 67, which illustrates a combustion chamber provided with a central four-hole sprayer, viewed in a direction normal to the plane of rotation of the air. If the speed of rotation of the air is too great, air used up by spray A will be carried into the path of spray B, and air from spray B into that of spray C, and so forth. In an extreme case the used-up air from A might cross the path of sprays C and D, or even that of A itself. At too low a swirl speed the injection may have ended before the whole of the available air has crossed the path of the sprays, and under these circumstances we are either not making use of all the air which is available or we are attempting to burn too much fuel in the air that we are using. It may be mentioned here that in the event of an excess of fuel being delivered to one portion of the air while another portion is left with a deficiency, the distribution of the fuel does not automatically right itself later on in the cycle. Some correction may perhaps take place, but only to a very limited extent,

except in the case of certain combustion chambers of a special type, which are referred to later.

If a high output is to be obtained from an engine, it is therefore essential that the movements of the air and of the fuel should be properly matched with one another. From fig. 67 it will be seen that the correct movement of the air in the type of chamber illustrated will be for the air to move from A to A′ during the period in which the fuel is being delivered to the engine. The whole of the quadrant of air allocated for each spray will then have passed through the zone of its own particular spray during the period when the fuel is looking for oxygen. If the chamber were fitted with two sprays only, these would of course be spaced at 180° apart and the rate of swirl would have to be a little over twice as great as with four sprays. The fact that the difference is not exactly two to one is that the angle between A, A′ is not actually 90° but something less, the difference depending upon the angle occupied by the spray. With two sprays the angle from A to A′ will be 180° minus the angle occupied by the spray, which in all probability will not be substantially different from that occupied when a four-hole sprayer is used.

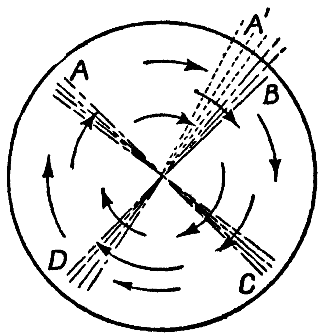


Fig. 67

It was explained earlier that the whole of the fuel burned in zones adjacent to the outer wall must pass through the zones nearer to the nozzle. This would appear to conflict somewhat with the statement just made that zones overcharged with fuel are not fully compensated for at a later stage in the cycle. Under operating conditions two factors help to mitigate this condition. The delay period ensures that quite an appreciable quantity of fuel travels some distance away from the nozzle before any combustion takes place. The delay period is not confined to the first part of the fuel to be delivered. The fuel which arrives after ignition has taken place is also subject to delay; it requires time, although somewhat less time than the first fuel to arrive, to reach its ignition point, even when injected into flame. Quite a considerable percentage of the total fuel charge thus passes through the zone near the nozzle without any combustion taking place. The second factor is that the fuel itself causes a movement of the air by carrying air along with it and thus causes a certain amount of renewal of the air in the immediate neighbourhood of the nozzle.

The fact that all the fuel travels some little distance away from the nozzle before ignition takes place plays quite an important part when the fuel is injected radially inwards from an injecter situated at the

perimeter of the plane of rotation. At first sight such an arrangement might be thought to be equivalent to a two-hole central sprayer. In practice, however, it is the equivalent of a single-hole sprayer and requires a swirl of the order of four times that required by a four-hole central sprayer. It would appear, therefore, that by far the largest part of the combustion takes place on the side of the centre of rotation remote from the nozzle—a supposition which is supported by measurements of the temperature of the combustion chamber walls adjacent to the nozzle. This of course applies to small chambers only.

4. Methods of Producing the Air Swirl.

The air swirl may be produced in several ways: (1) during induction; (2) during compression; (3) during combustion.

All three methods are employed by different designers, and in some instances the latter two are employed together.

5. Induction-induced Swirl.

Induction-induced swirl, as its name implies, is induced while the charge of air is entering the cylinder. It is produced by admitting the air in a tangential direction so that, guided by the cylinder wall, it assumes a rotary movement about the axis of the cylinder. The tangential direction is obtained in one of several ways: by masking one side of the inlet valve so that admission takes place around part of the periphery of the valve only and the air is thereby directed in the desired direction, as shown at I (fig. 68); by sloping the port itself in the desired direction, so that the air is guided in such a manner that the bulk of it is discharged from one side of the inlet valve (II, fig. 68); by providing a projecting lip over one side of the inlet valve seat (III, fig. 68); or, as in the case of the Ricardo sleeve-valve engines, by arranging the bars between the inlet ports at a suitable angle (IV, fig. 68). In the case of the single sleeve a swirling movement is produced even without a radial disposition of the ports by reason of admission starting at one corner of the port, * as shown at *b*, in IV.

The method most commonly employed is to mask the inlet valve. This has the great advantage of being readily adaptable if some modification of the swirl is found to be necessary. During development work the mask can be adjusted while the engine is actually running; the effect of any change can be noted immediately, and the optimum position is thus determined very quickly. The same procedure can be adopted with IV if hinged port bars are provided for the during development work. With II and III very little alteration is possible once a casting is made, so that one has to adjust the fuel injection

* Alcock, *Proc. I. Mech. E.*, Dec., 1934.

equipment to the swirl provided by the casting instead of being free to adjust both the swirl and the injection characteristics.

Arrangements such as that shown at II are usually put forward under the plea that the use of a masked valve entails a serious loss of breathing capacity and that the loss will be avoided by this arrangement. The mask must, of course, result in some loss of breathing capacity, but even when the mask occupies a quite large proportion of the perimeter of the valve this is nothing like as much as has been supposed, and at the speeds at which engines are operated to-day is almost negligible.

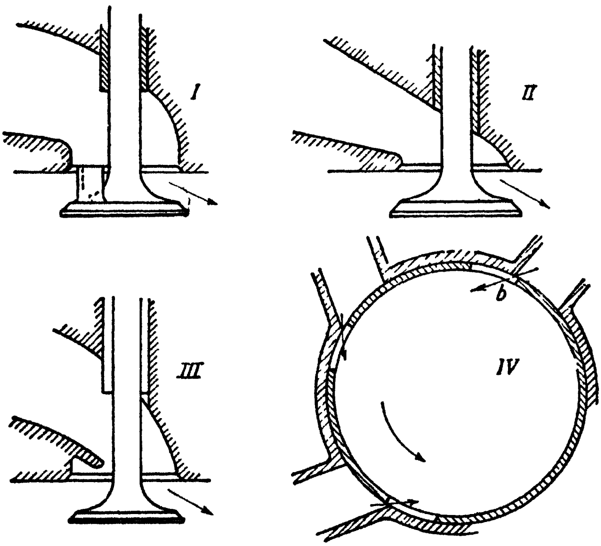


Fig. 68.—Methods of creating an induction-induced swirl

It will be realized on reflection that if the arrangement shown at II can be so made that the air flows only through the side of the valve opposite to that from which it is brought up to the valve, then the loss of useful circumference must be substantially the same as if the inoperative side of the valve were occupied by a mask.

The employment of a masked valve does, however, introduce a somewhat objectionable feature. It is necessary to prevent the valve from turning, so as to keep the mask always in the desired position. With a little ingenuity, however, this difficulty is easily overcome.

The swirl velocity within the cylinder comes primarily from the velocity of the air stream through the inlet valve. The actual speed of rotation can, however, be modified to quite a wide extent by the direction the air flow is made to take on entering the cylinder. The

speed of rotation attained will depend upon the velocity of the air through the valve and the distance of the centre of the flow from the centre of the cylinder.

Thus if the air stream enters the cylinder (fig. 69) in the direction AB at the velocity V ft. per min. and the distance of the centre of the stream is R_1 ft. from the centre of the cylinder, then the speed of rotation of the air around the cylinder will be a function of V/R_1. The actual value is not quite so simply determined, however, not only because of the expansion which takes place after passing through the valve but also because the air is directed in a somewhat downward direction, and further, under certain circumstances part of the air stream will pass on the side of the centre of the cylinder opposite to that on which the centre of the stream passes and will therefore have a retarding effect. It will be obvious, however, that the rate of swirl will vary greatly according to the direction relative to the centre of the cylinder in which the air stream enters. If we have a valve provided with a mask occupying a part of its circumference and place the mid-point of the mask on the line xO in fig. 69, which passes through the centre of both the valve and the cylinder, the mask itself lying against the cylinder wall, it will be clear that no rotational movement of the air about O will be produced, because the air stream will pass equally on either side of the centre; if the mask were placed in the exactly opposite position there would be no rotation, because two equal and opposite streams of air would again be produced. If now the valve is turned through a small angle to bring the centre of the air stream to one side of the centre of the cylinder, the stream of air acting upon the gases already in the cylinder will cause them to assume a rotary movement. The stream of air striking against the cylinder wall will be deflected and assist in creating a rotary movement in the same direction.

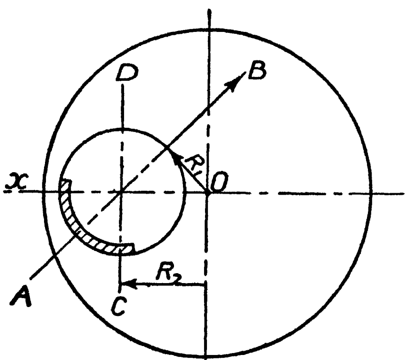


Fig. 69

With a small angular displacement of the mask from the position on xO a considerable portion of the air stream will pass the centre of the cylinder on the side opposite to that upon which the centre, and therefore the bulk, of the stream is passing. This part of the air stream will have a retarding effect and limit the speed of rotation. As the

angular displacement of the mask is increased, this retardation influence is reduced, and the speed of rotation for a given air stream speed will increase and will ultimately reach a maximum, to be followed by a decline as a further rotation of the valve provides a second opening through which a stream of air will issue on the opposite side of the centre of the cylinder and thus introduce a retarding effect (fig. 70).

The proportion of the circumference of the valve which is masked has a direct bearing upon the swirl which is produced. As the swirl comes primarily from the velocity of the air as it enters the cylinder, and as the velocity of the air depends on the rate of flow and the area of the aperture through which it flows, it follows that for a given rate of flow and a given direction of flow the smaller the aperture the greater will be the swirl. The object of the designer should be to produce the swirl necessary for the desired performance with the minimum of restriction to the air flow, i.e. with the maximum possible volumetric efficiency. There is nothing to be gained, and much may be lost, by providing an arrangement of mask which is capable of producing more swirl than is required and then "detuning" the set-up to produce just the necessary amount of swirl. The ability to produce excess swirl means that the resistance of the induction system has been increased more than is necessary, and that the volumetric efficiency has therefore suffered. The extent of the reduction in breathing capacity may be small, but it is obviously bad policy to add even the smallest unnecessary resistance. Detuning of the system will often mean that the position of the mask for optimum results is sharply critical and that even a small departure from this position causes a serious loss of performance. Such a condition is highly undesirable.

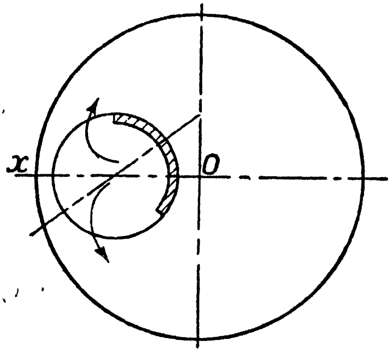


Fig. 70

A further, and even more serious, objection is that with such an arrangement it is frequently found that the optimum position of the mask differs with the speed of the engine. This may not matter a great deal when the engine is intended to operate at a fixed speed or is required to deliver its maximum output at one fixed speed only, as in the case of engines coupled to a screw propeller, but for an engine intended to give its full torque at a number of different speeds it is a very serious defect. The arrangement of the mask should be such that the engine is able to give its best performance at each and every speed at which it is designed to operate. Failing this—and it may not always

be possible to attain this ideal condition with every design, especially if the speed range is unusually wide—matters should be adjusted to give the optimum results over the most important part of the speed range.

During development work it is highly desirable to arrange the inlet valve so that the position of the mask can be moved easily around a full circle while the engine is running. If this is done, and it is quite simple to arrange, a great deal of time and trouble will be saved. The optimum position of any mask can be quickly and accurately determined by revolving the valve slowly while the engine is under power and noting the changes in dynamometer pull and exhaust colour.

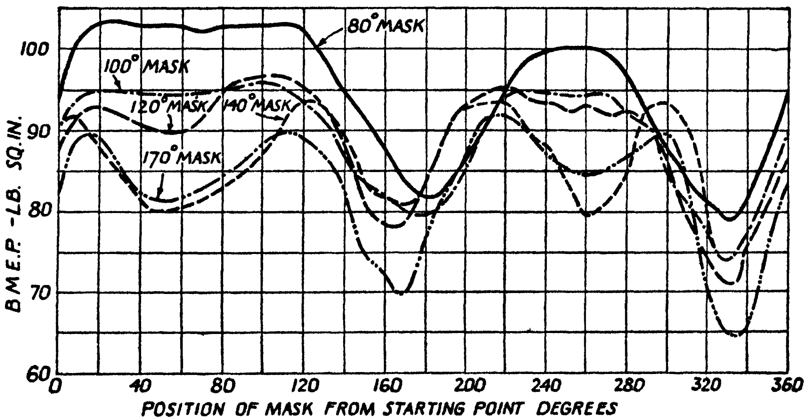


Fig. 71.—Influence of mask size and position on B.M.E.P. at 1000 r.p.m.

If a direct reading of the actual mask position can be obtained, it is better still, because the results of the experiment can be plotted and the relationship between mask position and performance can be studied. A test carried out with a number of masks of different size will quickly enable the best all-round combination to be determined.

The results from an experiment such as this are shown in figs. 71 and 72, which give a series of curves obtained at speeds of 1000 and 1500 r.p.m. from a single-cylinder experimental unit fitted with a four-hole sprayer. For the purposes of this experiment the fuel pump was set to an arbitrarily selected delivery, the inlet valve was rotated through a full circle, and readings were taken at steps of 10°. The speed was kept constant and the pull on the dynamometer and the colour of the exhaust were noted. The engine was tested with masks extending for 170°, 140°, 120°, 100° and 80° around the circumference. It should be noted that the fuel delivery was not necessarily exactly the same with each mask, so that it is the relative form of the B.M.E.P.

curves which is of interest rather than the actual values obtained with the different masks at any given mask position. It will be observed that starting from the zero swirl position the B.M.E.P. rises rapidly as the mask is moved into a swirl-producing position, but that with masks which extend for the greater distances around the valve the B.M.E.P. after reaching a sharp peak rapidly falls away again, to rise once more to a peak before falling to a low value as the second zero swirl point is reached. The process is repeated as the second zero swirl point is passed and a swirl in the reverse direction of rotation is produced.

As the extent of the mask is reduced, the depth of this cusp decreases and finally disappears, giving a broad flat-topped curve. The

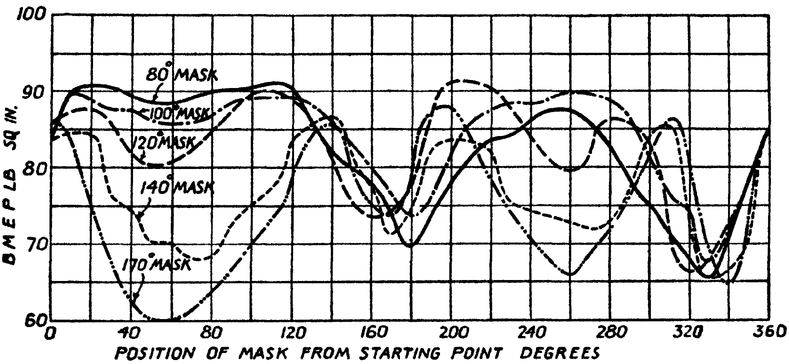


Fig. 72.—Influence of mask size and position on B.M.E.P. at 1500 r.p.m.

adoption of a mask occupying a big angle would have necessitated the valve being set very accurately in position if the best performance was to be obtained at any one speed, but the peak positions for the two speeds do not coincide any too well. With the 80° and the 100° masks, however, a wide flat-topped curve is obtained allowing considerable latitude in the selection of position.

The fall-away which takes place with the large angle masks is due to overswirling causing the products of combustion of one spray to be brought into line with the one following it before combustion has been completed. This condition might possibly be corrected by adopting a higher rate of injection, but this would tend to increase the maximum pressure, the rate of pressure rise, and also the quantity of fuel delivered during the delay period, and so tend to produce rough running. Also, a change in the rate of injection would not be likely to correct the lack of coincidence between the peaks at the different speeds.

Overswirling is more likely to arise when multi-hole sprayers are

used. The greater the number of sprays the more easy it is to produce an excessive rate of swirl. Where a single-hole sprayer is used the problem is more often that of how to produce sufficient swirl to enable the optimum results to be obtained.

It has already been stated that for the optimum results the swirl and the injection characteristics must be accurately matched with one another. The function of the swirl is to bring the air to the fuel so that the correct supply of air is available as and when required, but at the present state of knowledge this can only be done by actual experiment and that very largely of the "cut and try" variety. It is, however, very doubtful whether it can ever be anything else but a matter of experiment, because the number of variables is so great as to make it almost impossible to produce an expression, or series of expressions, incorporating all the factors necessary to provide the correct solution. This does not mean that when dealing with engines of similar design but of different cylinder capacity, the experience gained with one size cannot be applied with some measure of success to another, but even then a fair amount of experiment will certainly be necessary in order to obtain the best results, unless a very lucky shot has been made.

6. The Measurement of Air Swirl.

The measurement of the actual swirl within the engine cylinder is a matter presenting considerable practical difficulties. A number of attempts have been made, mostly on engines of larger sizes than are commonly associated with high speeds, and a number of the methods used in these attempts are mentioned in a paper by J. F. Alcock and in the discussion that followed.* For obvious reasons most of the attempts have been made when the engine is being motored round or is driven by the remaining cylinders while the one under observation has the fuel cut off. Probably the most successful attempts are those made by Ricardo on his sleeve-valve engine, an engine which lends itself admirably to such an experiment. By the use of a light aluminium vane lying across the combustion chamber and connected to a counting gear by means of a spindle passing through an almost frictionless labyrinth packing, he was able to correlate swirl speed and engine performance by assuming that an engine when operating normally had the same rate of swirl as that obtained when it was motored with the anemometer in position. These experiments showed that for the type of engine used for the experiments a swirl speed equal to ten times the crankshaft speed was required for the optimum results, the performance improving rapidly as this speed was approached and falling off at a similar rate as the ratio of 10 : 1 was exceeded.

* Air Swirl in Oil Engines, *Proc. I. Mech. E.*, Dec., 1934.

The results obtained are shown in figs. 73 and 74, which are taken from the paper by Alcock referred to above. Fig. 73 shows the effect of the rate of swirl velocity upon the maximum B.M.E.P. obtainable, while fig. 74 shows how the influence of rate of swirl affects the engine under part-load conditions. It is clear that not only does swirl speed affect the maximum smoke-free B.M.E.P. and the fuel consumption

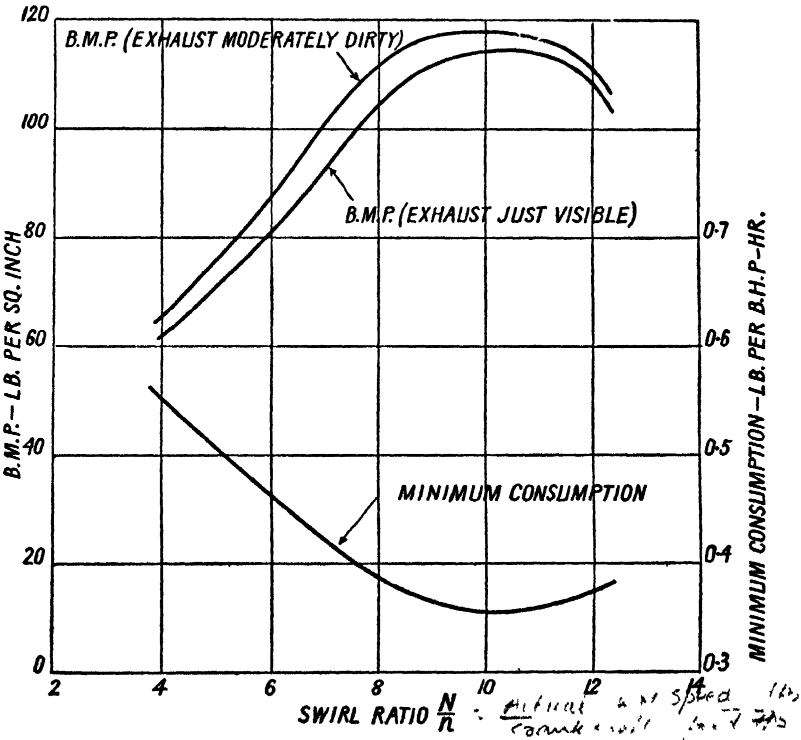


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Fig. 73.—Influence of swirl upon power and fuel consumption in a Ricardo "Vortex" engine running at 1300 r.p.m., with constant injection timing

at that B.M.E.P., but it also affects the fuel consumption at all loads throughout the range. This shows that it is not only a question of arranging the air flow to suit the maximum load conditions, but of arranging it so that combustion is cleanly and quickly completed at all loads. With a swirl rate too slow for the rate of injection, too much fuel will be added to that part of the air which passes in front of the nozzle during the injection period and the combustion conditions will be unsatisfactory, although the total quantity of air available may be ample to consume the whole of the fuel charge if the two are brought

together properly. Apart from this, a low rate of swirl will result in a slow rate of burning, with a corresponding loss in efficiency. The low rate of burning is well illustrated by fig. 75, which is also taken from Alcock's paper.

The particular engine used in Ricardo's experiments was provided with a single-hole nozzle which discharged the fuel near the circumference of the combustion chamber and in a direction parallel to the axis of rotation of the air. This arrangement, while it calls for a simple form of nozzle, calls also for a high speed of rotation if the whole of the air is to be brought across the path of the fuel during the injection

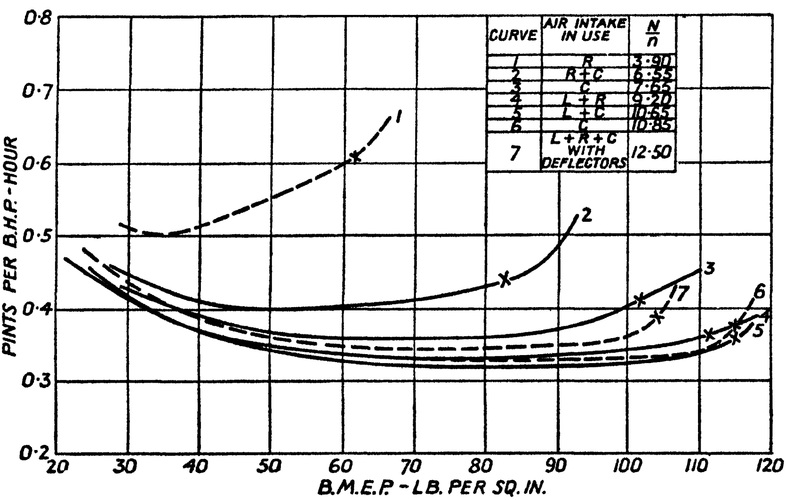

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Fig. 74.—Influence of swirl at reduced engine output

period. The optimum rate of swirl is partly dependent upon the duration of the injection period, but as, for reasons of economy, the combustion must be completed shortly after expansion begins and cannot be allowed to start more than a few degrees before top dead centre, the duration of injection under full-load conditions is confined within comparatively narrow limits, and the latitude in swirl ratio is thus correspondingly restricted. A swirl ratio of 10 : 1 means that the air passes across the face of the nozzle during a period of 36° of crankshaft rotation. The spray, however, occupies a definite arc of the circumference of the chamber and so reduces the effective distance the air has to travel, or rather, reduces the possible injection period to something less than 36° of crankshaft rotation.

A swirl ratio of 10 : 1 cannot of course hold for all types of engines, as it will obviously be modified by the number of fuel sprays. The

duration of injection will not vary greatly from engine to engine for reasons just given above, so that the swirl ratio necessary for any engine with more than one spray may well be taken, roughly, as $10/n$, where n is the number of sprays. Actually there will be some departure from the absolute value of $10/n$, which will be governed by the dispersal of the sprays, in addition to any modifications introduced by such differences as may exist in injection rates and any other factors which may influence the results.

In one particular engine which gave good results the injection period was found to be 20° of crank rotation at the normal full load; the nozzle was provided with four spray holes and, ignoring any question

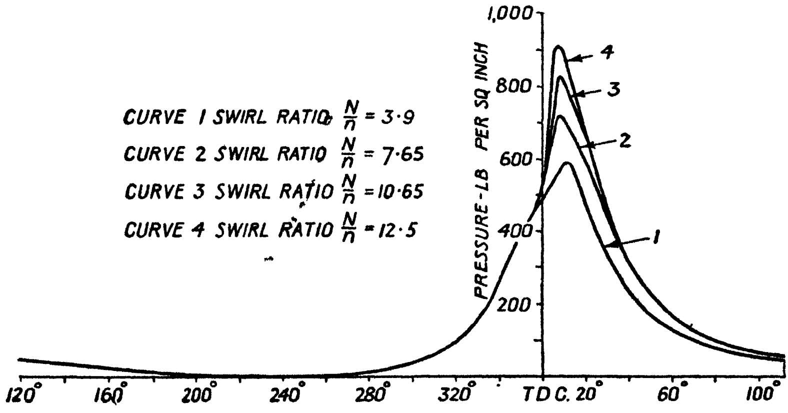

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Fig. 75.—Influence of swirl upon the rate of combustion

of spray dispersal, the air would be required to make one-quarter revolution, 90°, in 20° of crankshaft rotation, giving a swirl ratio of 4·5 : 1. If, however, we allow, say, 15° for spray dispersal, we have the air turning through 75° for 20° of the crankshaft, giving a swirl ratio of $75/20 = 3{\cdot}75 : 1$.

In the case of an engine fitted with poppet valves it is, unfortunately, not possible to make a direct measurement of the swirl in the way Ricardo has done, that is, not with the size of cylinder usually associated with speeds above about 1000 or 1200 r.p.m., because the small depth of the combustion chamber and the projection of the valves when open preclude the presence of an anemometer vane of adequate proportions. Although the direct measurement of swirl may not be possible, however, much useful information as to the nature and extent of the swirl which can be produced by means of masked valves may be obtained by flowing low-pressure air through the

induction system and measuring the swirl produced in the cylinder after the piston has been removed and an anemometer fitted in the bore. This anemometer must, of course, be fitted with a perfectly flat vane which will respond to a rotary movement of the air only.

A set of curves of swirl speed against mask position obtained in this way are shown in fig. 76. These were obtained from the same cylinder and head as the curves shown in figs. 71 and 72, and the valve having the mask extending for 140° around the circumference was used. Measurements were made of the rate of swirl produced for a

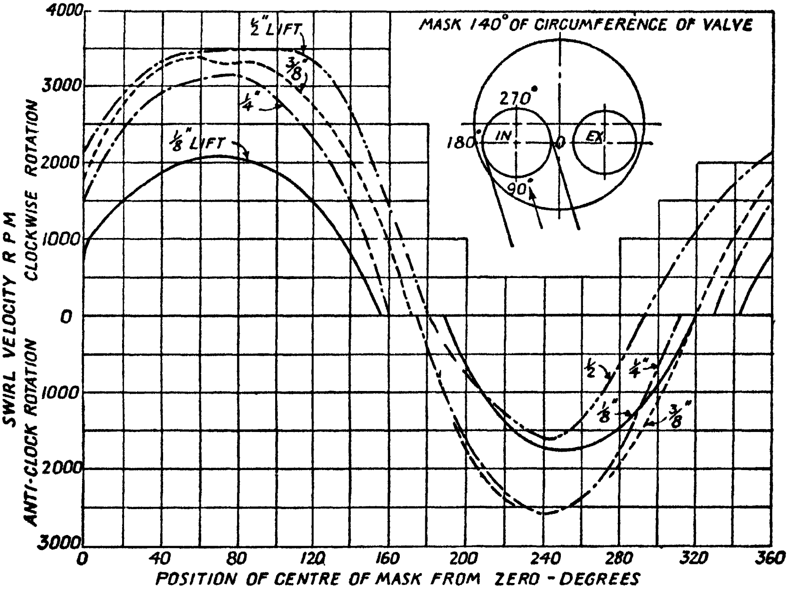

Fig. 76.—Swirl velocity produced by a pressure drop of 1 in. of mercury across the valve at different valve lifts and mask positions

pressure drop across the valve of 1 in. of mercury up to a lift of ½ in. in steps of ⅛ in. It will be noted that the swirl velocity in a clockwise direction is materially greater than that obtained in the anti-clockwise direction. This can only be explained by the fact that the direction in which the air arrives through the port has some influence on the swirl. With an anti-clockwise rotation the air was turned back on itself and entered the cylinder in a direction opposite to that in which it was travelling through the inlet port.

It should be pointed out that although the pressure drop across the valve is kept constant at all positions of the mask this does not necessarily mean that the weight of air passing in unit time is the same at each position. The true extent of the masking will usually be found to

vary considerably with the position of the mask. In practically every engine the provision of valves of adequate diameter brings the head of the valve in close proximity to the cylinder walls at one point and thus permanently masks part of the circumference of the valve. The position of the mask relative to this point may make a considerable difference to the actual length of the masked circumference of the valve, as is illustrated in fig. 77. This shows a valve which is provided with a mask extending for 90° around the circumference, but when placed in a certain position this can be made to produce the effect of a mask occupying nearly twice this angle. The effective area of the valve opening thus changes as the position of the mask is changed, and the discharge for a given pressure drop will change also. Some

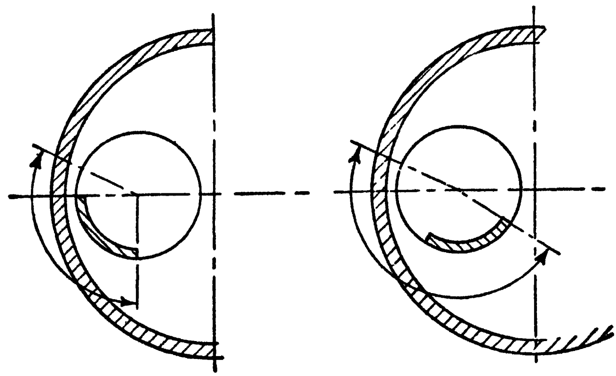

Fig. 77

idea of the extent of the change in the discharge will be gained from fig. 78, which gives the figures obtained from the same valve as was used for the experiment illustrated in fig. 76.

If the results of the experiments illustrated in figs. 71 and 72 are compared with those in fig. 76, it will be seen that the mask positions at which the four peaks were obtained with the valve having a mask angle of 140° correspond to four positions at which the mask gives substantially the same swirl speed in the blowing experiments. This and the fact that the same B.M.E.P. is reached at each of the peaks are as would be expected. The fact that exact equality is not attained is probably attributable to differences in breathing capacity at the several positions.

Blowing experiments for the measurement of swirl are greatly increased in value if the weight of air is measured at the same time. This can easily be done by means of an orifice, the degree of accuracy being ample for the purpose in hand. The chief problem in most cases is that of obtaining an adequate supply of air, but this can be taken

from the shop supply of compressed air reduced to a suitably low pressure. In cases where the supply is inadequate a convenient method is to connect the blowing model to the air intake of an engine on the test stand. The air flow may then be varied at will by adjusting the speed of the engine. One point to remember is that since the suction stroke occupies only one-fourth of the time of the complete cycle, for comparable rates of flow the quantity of air flowing through the model must be four times that passing through the engine valve in the same space of time.

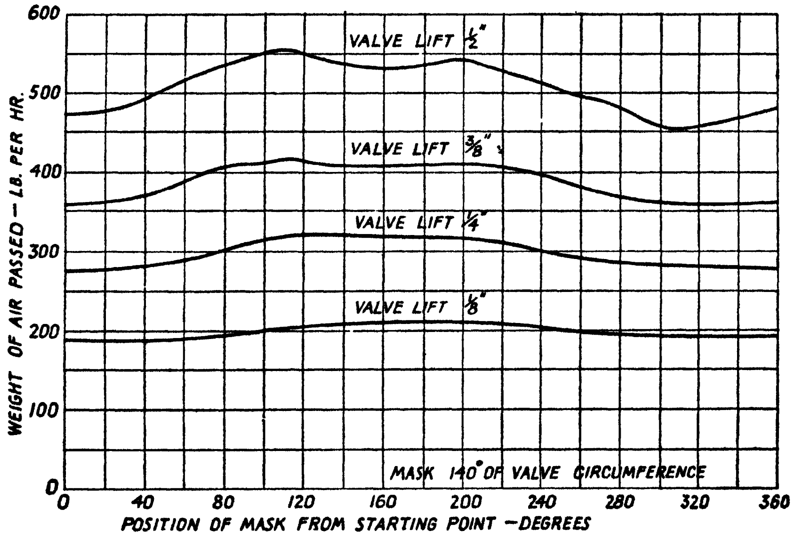

Fig. 78.—Variation in air flow with change in mask position. Pressure drop across valve 1 in. of mercury (for mask position see fig. 76)

A point which the blowing experiment brings out is the very low efficiency of the conversion of inlet velocity into swirl. To quote an instance: a valve with a 140° mask at a lift of ¼ in. gave a maximum swirl speed of 3120 r.p.m. The quantity of air flowing through the valve was 304 lb./hour at a temperature of 7° C. and a barometer of 759 mm. The diameter of the throat of the valve was 1·73 in., so that the area of opening was

$$\pi \times 1{\cdot}73 \times \frac{220}{360} \times {\cdot}25 = {\cdot}83 \text{ sq. in.}$$

The volume of air passing was

$$12{\cdot}4 \times \frac{304}{60} \times \frac{280}{273} \times \frac{760}{759} = 64{\cdot}5 \text{ c. ft./min.,}$$

and the velocity through the opening was

$$\frac{64{\cdot}5 \times 144}{{\cdot}83} = 11{,}200 \text{ f.p.m.}$$

The diameter of the cylinder was 4·53 in. (115 mm.), and the centre of the valve was 1¼ in. from the centre of the cylinder. If we were to assume that a velocity of 11,200 f.p.m. was produced at a radius of 1¼ in. from the centre of the cylinder, the rotational speed would be

$$\frac{11{,}200 \times 12}{2\pi \times 1{\cdot}25} = 17{,}100 \text{ r.p.m.}$$

The actual figure of 3120 thus represents an efficiency of only 3120/17,100 = 18·2 per cent.

Even if we assume that a speed of 11,200 f.p.m. should be taken at the circumference of the cylinder the speed of rotation would be

$$\frac{11{,}200 \times 12 \times 2}{2\pi \times 4{\cdot}53} = 9400 \text{ r.p.m.},$$

representing an efficiency of only just over 33 per cent. The latter figure is obviously higher than the correct efficiency, and the former is probably something less than the truth.

This loss of efficiency is to be explained by eddying and friction, and the fact that the mask does not entirely prevent air from entering on the masked side. The air entering in a single stream partakes of the nature of a jet of gas discharging into space, a condition which involves a very great loss of energy. If the air could be divided up amongst a fairly large number of jets arranged around the circumference of the cylinder the swirl efficiency would be materially increased. This is proved by the much greater rate of swirl which can be produced by using a sleeve valve which enables the air to be delivered in several streams. A swirl rate several times as great as that which can be produced by a masked poppet valve can be produced by the sleeve valve, the swirl rate being great enough to enable a single-hole sprayer to be used without the necessity of spring-loading the orifice. Such a combination does not appear to be possible with a masked valve—at any rate with a masked valve capable of giving adequate breathing capacity.

7. Increasing Induction-induced Swirl.

Induction-induced swirl is frequently augmented by transferring the air during the compression stroke to a combustion chamber which has a diameter somewhat less than that of the cylinder itself. By this means the velocity of rotation of the air is increased beyond that which it possessed while it remained in the cylinder itself and, at the

same time, the air is made to assume a more compact form. In forcing the air inwards against centrifugal force, work is done which appears as an increase in velocity greater than that which would result solely from the fact that the air is rotating in a smaller circular path. By the principle of the conservation of the moment of momentum,

$$I_1\omega_1 = I_2\omega_2,$$

where I_1 and I_2 are the moments of inertia of the air when in the cylinder, and after having been transferred to the combustion recess, and ω_1 and ω_2 are the corresponding angular velocities.

But $I = mk^2$, where m is the mass of air and k its radius of gyration, so that

$$\omega_2 = \frac{I_1\omega_1}{I_2} = \frac{mk_1^2\omega_1}{mk_2^2} = \frac{k_1^2}{k_2^2}\omega_1 = \frac{r_1^2}{r_2^2}\omega_1.$$

The angular velocity is therefore inversely proportional to the square of the radius of gyration, and hence inversely proportional to the square of the radius of the chamber. This of course assumes that no loss of any kind is sustained during the transfer of the air from one chamber to the other, a condition not fulfilled under actual working conditions.

Similarly, it can be shown that the kinetic energy contained in the mass of gas when transferred to the reduced diameter will be

$$K_2 = \frac{K_1 \times I_1}{I_2} = \frac{K_1 \times r_1^2}{r_2^2},$$

again assuming that no loss occurs during the transfer.

This increase in kinetic energy has to be obtained from somewhere, and the source obviously must be the pressure used to force the gases inwards against the centrifugal force.

That a loss of considerable magnitude is likely to be incurred during the transference of the air from one diameter to another may well be imagined, because the transference will not be effected without a good deal of eddying, since the first part of air to be transferred will be rotating in contact with, but at a higher rate than, the air remaining in the cylinder. During the earlier stages of compression this may not amount to a great deal, but during the final stages, when the density is high, the effect must increase considerably. Very little information is available as to the actual relationship between the swirl produced in the cylinder during induction and that produced in the combustion chamber. Fig. 79 shows some results given by Alcock (*loc. cit.*) which were obtained from a sleeve-valve engine tested with three different diameters of combustion chamber—5·5 in., the full diameter of the cylinder, 4·035 in. and 2·6 in. The swirl in this instance

varied very nearly inversely as the diameter. The diagram also indicates the advantage of using a combustion chamber of small diameter

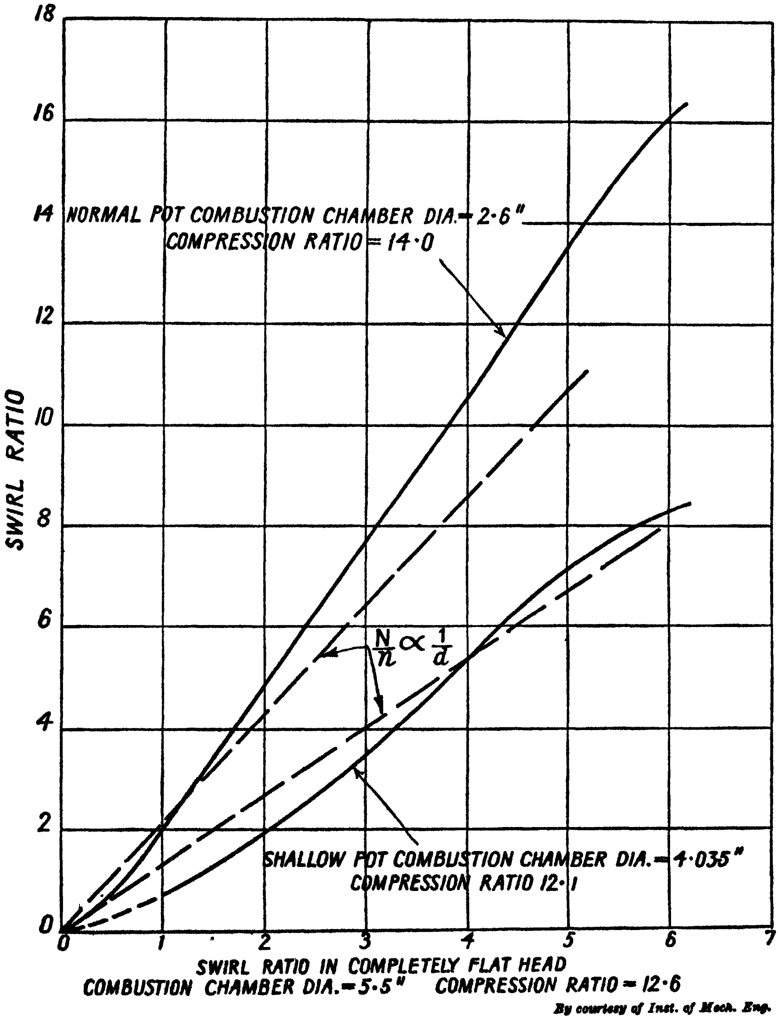

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Fig. 79.—Showing the influence of combustion chamber diameter upon swirl ratio

relative to that of the cylinder, the increase in swirl in this instance being rather higher proportionately than with the larger diameter.

The combustion chamber may be placed in either the cylinder head or piston crown as circumstances dictate, but in order that the damping effect during the transfer may be a minimum, the chamber

must obviously be placed practically coaxial with the cylinder. Some small departure from the truly axial position may be permitted, but any very great departure will result in increased fluid friction between the gas in the two chambers, with a consequent loss of energy. This condition practically compels the designer to place the combustion chamber in the piston when poppet valves are used, because of the difficulty in accommodating valves of adequate size if the combustion recess is placed centrally in the cylinder head. With a sleeve valve, as in the Ricardo " Vortex " engine, the combustion recess may advantageously be placed in the cylinder head.

The ratio of diameter of cylinder to diameter of combustion chamber does not seem to be governed by any fundamental requirements of the engine. The swirl ratio, i.e. the ratio between the rate of air swirl and the rotational speed of the engine, varies in different designs, being governed largely by the number of fuel sprays or the form of spray used and the rate of injection, and for any given engine it will be clear that the required swirl ratio can be obtained from numerous differing combinations of inlet swirl velocity and the ratio of combustion recess and cylinder diameters.

The actual chamber diameter will depend to some extent, and, it would appear, to a quite large extent, upon the spray penetration. This being so, for any given cylinder diameter the ratio of cylinder and chamber diameters is settled automatically and the requisite swirl ratio must be obtained by adopting a suitable initial swirl velocity. A change in fuel nozzle characteristics may necessitate a modification of the diameter of the chamber and a readjustment all round, and hence it is not possible to lay down rules for ascertaining the required proportions, which must be determined by experiment for each particular case.

The determination of suitable conditions is rendered much easier than the foregoing may suggest because of the ease with which the initial swirl velocity may be altered when masked valves are used. By the simple expedient of arranging the inlet valve so that it can be rotated while the engine is running, the optimum position for that particular valve is very quickly determined, and it does not take very long to repeat the experiment at a number of different speeds and with a number of valves having masks of differing dimensions, and thus arrive at the one which gives the best all round results.

The determination of the optimum diameter for the combustion recess is a much more lengthy process, and requires tests with different heads or pistons. A change in chamber diameter influences not only the penetration but also the " squish " effects (see below); this is an illustration of the difficulty of altering one feature without changing another one at the same time.

8. Secondary Air Movement or " Squish ".

In the case of engines fitted with the open type of combustion chamber, the combustion chamber itself is almost invariably somewhat smaller in diameter than the cylinder, and is formed by a recess provided in the piston or cylinder head. In order to concentrate as much of the air as possible within this combustion recess it is usual to bring the piston to within a very small distance of the cylinder head at the top dead centre position. The minimum distance varies with different designs, but figures as small as ·06 in. and even ·04 in. are common, and even smaller clearances have been used, although these call for special precautions in machining and assembly.

The reduction of the clearance to such small dimensions produces an important secondary movement of the air, the extent and importance of which have not always been fully appreciated. Its effect is to produce a flow of the air radially inwards towards the combustion recess by squeezing it out from between the piston and the cylinder head as they approach each other at the end of the stroke. This movement, termed " squish " by Ricardo, will be added to any other movement which may already exist, and as it makes its appearance and is most vigorous just about the time when injection is timed to begin, it can have a very pronounced effect upon the behaviour of the engine.

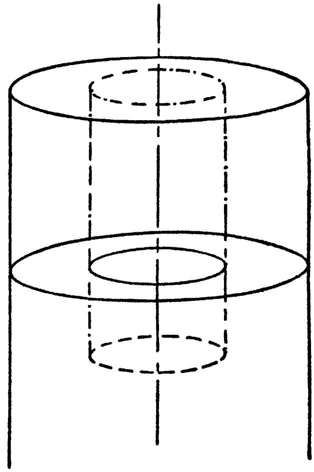


Fig. 80

Strictly speaking, squish is a form of compression-induced air movement, but as its use is associated with engines utilizing induction-induced swirl it may conveniently be considered under that heading.

Considered by itself, the action of the squish is simple. The air within the cylinder may be considered as being divided into two parts, (*a*) an inner cylindrical portion with diameter equal to that of the opening into the combustion recess and length equal to the distance between the top of the piston and the cylinder head plus the depth of the combustion recess, and (*b*) an annular portion which surrounds the other and lies between the piston top and the cylinder head, as shown diagrammatically in fig. 80.

As the piston approaches the cylinder head the total volume of air is reduced, but at the same time the proportion of air contained within the inner cylindrical portion gets greater and greater because

of the fixed volume represented by the combustion recess. There is thus a transfer of air from the annular portion to the cylindrical portion, the air from the annulus being squeezed out and flowing radially inwards into the cylindrical portion. The streams from opposite sides of the annulus meet one another, and are therefore deflected downwards (or upwards, according to the location of the combustion chamber) into the combustion chamber, and on reaching the end of the chamber will flow radially outwards towards the outer walls, up which they will again be deflected towards the open end. Here they will be met by the air flowing radially inwards from the annulus, to be carried round again, producing a toroidal movement within the combustion chamber (fig. 81).

At first sight it might appear that because the velocity of the piston is falling rapidly to zero as top dead centre is approached, the movement obtained from the squish could not be of sufficient intensity to have any real value. Actually, however, the ratio of the volume of air in the annulus to that contained in the remaining portion is changing very rapidly towards top dead centre. At top dead centre the volume of the annulus is reduced to that given by the mechanical clearance allowed between the piston and the cylinder head, and the rate of change of the volume of the annulus is high as top dead centre is approached.

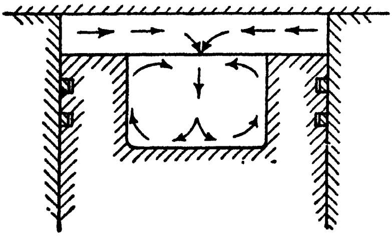


Fig. 81

During the final stages of compression the density of the air is high, so that even if the velocity of flow as measured in feet per second may not have a very high value, the energy represented is considerable. Actually, it will be found that in cases where the difference in diameter between the combustion chamber opening and that of the cylinder is relatively large, the rate of flow from the annulus has its maximum value quite close to top dead centre.

In view of the very real importance of the squish effect it is worth while considering the actual extent of the squish velocity. Its value will depend both upon the minimum clearance between the piston and the cylinder head and the ratio of the diameter of the opening into the combustion chamber to that of the cylinder. It will depend, also, upon the compression ratio. Other things being equal, the higher the compression ratio the greater will be the proportion of air in the annulus at points near top dead centre, and hence the greater the rate at which it is expelled as the piston approaches the end of its travel. Broadly speaking, the squish velocity will vary directly with engine speed, although some departure from the strict

relationship will certainly be produced by the compressibility of the gas.

As the mass of the gas remains constant while its volume decreases, it is not possible to arrive at a solution by determining for any given point the instantaneous velocity of the piston and also the area of the exit from the annulus and so finding the rate of discharge from the annulus. Part of the gas which otherwise would have been discharged from the annulus by a given movement of the piston has remained in the annulus owing to the increase in density produced by the piston movement. It would, of course, be possible to derive a mathematical expression for this, but a step-by-step method is a simple means of arriving at a solution; it involves the use of nothing but simple mathematics and at the same time gives a very clear conception of what is taking place.

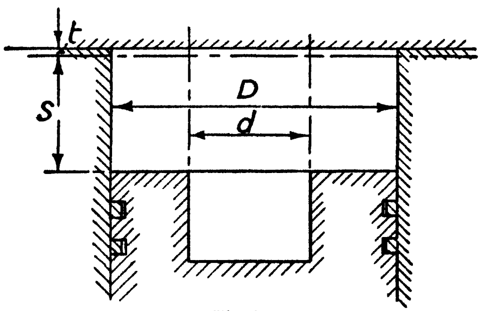


Fig. 82

Fig. 82 shows diagrammatically the cylinder of an engine in which D is the diameter of the cylinder itself, and d the diameter of the opening into the combustion chamber. S is the distance of the piston from its inner dead centre position and t is the clearance between the piston and cylinder head at top dead centre. To use the step-by-step method the density of the gas at each step must be determined in order to arrive at the relative proportions of the gas in the annulus and the remainder of the volume. The weight of gas flowing out of the annulus between successive steps can then be determined, and from this and the mean value of the density of the gas between the two steps the average velocity between the steps can be calculated. The method is, perhaps, a trifle tedious, but it is simple and it is not necessary to examine more than the final 45° of the compression stroke.

The gas is considered as being divided into two volumes, the annulus, which has a volume of $(\pi/4)(D^2 - d^2)(S + t)$ and the remainder, which has a volume equal to $(\pi d^2/4)(S + t) + V$, where V is the volume of the combustion recess in the piston (or cylinder head). The area through which the displaced gas flows from the annulus will be $\pi d(S + t)$.

The following example from an actual engine will serve to illustrate the method, and also the magnitude of the squish velocity.

Engine speed	1500 r.p.m.
Swept volume of cylinder	90 c. in.
Total volume of cylinder	96 c. in.
Compression ratio, total	16·0 : 1.
Compression ratio, effective	14·8 : 1.
Cylinder diameter	4·53 in.
Stroke	5·59 in.
Diameter of opening into compression space ..	2·48 in.
Clearance between piston and cylinder head at top dead centre	0·040 in.
Temperature of gases at beginning of compression (effective)	340° abs. (giving a density of ·0648 lb./c. ft.).
Volume of combustion recess in piston	5·37 c. in.
Connecting rod/crank ratio	4.

The movement of the piston with respect to crankshaft rotation is determined by the usual equation and is taken for 5° intervals (1/1800 sec.) for 45° from top dead centre.

In order to avoid dealing with unnecessarily small quantities, it is convenient to consider the air outside the annulus. The air outside the annulus is made up of the volume inside the combustion recess together with the cylinder represented by the opening into the recess (fig. 82) and the distance between the piston and cylinder head $(S + t)$.

The results from an analysis of the above engine are given in Table XXI, which perhaps requires a little explanation. Column 2 is obtained from the usual equation connecting crankshaft rotation and piston travel. Column 3 is, of course, column 2 plus the top dead centre piston-cylinder head clearance. Column 4 gives the total cylinder volume at angle $\theta°$ from T.D.C., while Column 5 gives the compression ratio at $\theta°$ and Column 6 the corresponding air density. Columns 7, 8 and 9 give the volume and weight of gas outside the annulus. Column 10 gives the change which has taken place in the weight of gas outside the annulus between successive 5° intervals. Column 11 gives the flow from the annulus in lb./sec., and is obtained by dividing Column 10 by the time interval for 5° rotation, i.e. 1/1800 sec. Column 12 gives the mean area of the exit from the annulus, which is given by $\pi d(S + t)$. The true mean is best obtained by plotting $S + t$ against θ and reading off from the curve. Column 13 gives the velocity of flow in ft./sec., and is obtained by multiplying the flow in lb./sec. by 144 and dividing by the product of the density in lb./c. ft. and the area of the exit from the orifice. Column 14, the pressure head required to create the flow in Column 13, is obtained from the usual equation $V = \sqrt{2gh}$.

TABLE XXI

1	2	3	4	5	6	7
Angle θ deg. from T.D.C.	Piston Distance		Total clearance volume c. in.	Compress. ratio at θ	Density at θ lb./cu. ft.	Vol. above combust. chamber opening c in.
	From T.D.C. in.	From Cyl. Head in.				
0	0·000	0·040	6·00	14·8	0·9590	0·193
5	0·014	0·054	6·225	14·26	0·9244	0·261
10	0·053	0·093	6·853	12·94	0·8399	0·449
15	0·120	0·160	7·932	11·21	0·7256	0·773
20	0·210	0·250	9·381	9·46	0·6135	1·207
25	0·324	0·364	11·216	7·91	0·5132	1·758
30	0·461	0·501	13·422	6·61	0·4288	2·420
35	0·621	0·661	15·998	5·55	0·3596	3·193
40	0·800	0·840	18·880	4·71	0·3048	4·057
45	0·996	1·036	22·035	4·02	0·2611	5·004

8	9	10	11	12	13	14
Total Vol. outside annulus c. in.	Weight of charge outside annulus lb.	Diff. in weight lb.	Mass flow lb./sec. at 1500 r.p.m.	Mean area exit from annulus sq. in.	Velocity of flow ft./sec.	Press. head equiv. to vel lb./sq. in.
5·549	·0030849	·000080	·14400	·3585	60·9	0·389
5·617	·0030049	·0001836	·33048	·5535	95·8	0·887
5·805	·0028213	·0002478	·44606	·966	86·4	0·628
6·129	·0025735	·0002410	·43380	1·583	59·7	0·256
6·563	·0023325	·0002199	·39562	2·345	43·2	0·113
7·114	·0021126	·0001821	·32778	3·420	29·4	0·044
7·776	·0019305	·0001513	·27234	4·54	22·3	—
8·549	·0017792	·0001190	·21420	5·87	15·9	—
9·413	·0016602	·0000945	·17010	7·31	11·9	—
10·360	·0015657					

These figures serve to show what a very real factor the squish velocity can become when the minimum clearance between the piston and cylinder head is made small and there is a fairly large difference in diameter between the combustion chamber opening and the cylinder diameter. They also bring out how much influence a change in cylinder head gasket thickness may have by altering the minimum piston-cylinder head clearance. The figures in Columns 11 and 13 are plotted in fig. 83 (p. 176), which gives also the corresponding figures if the minimum clearance is made 0·06 in. and 0·02 in.

The figures for the different clearances shown in fig. 83 are based upon the assumption that the compression ratio has not been changed by the alteration in clearance. In practice a change in the piston cylinder head clearance is frequently made without any adjustment

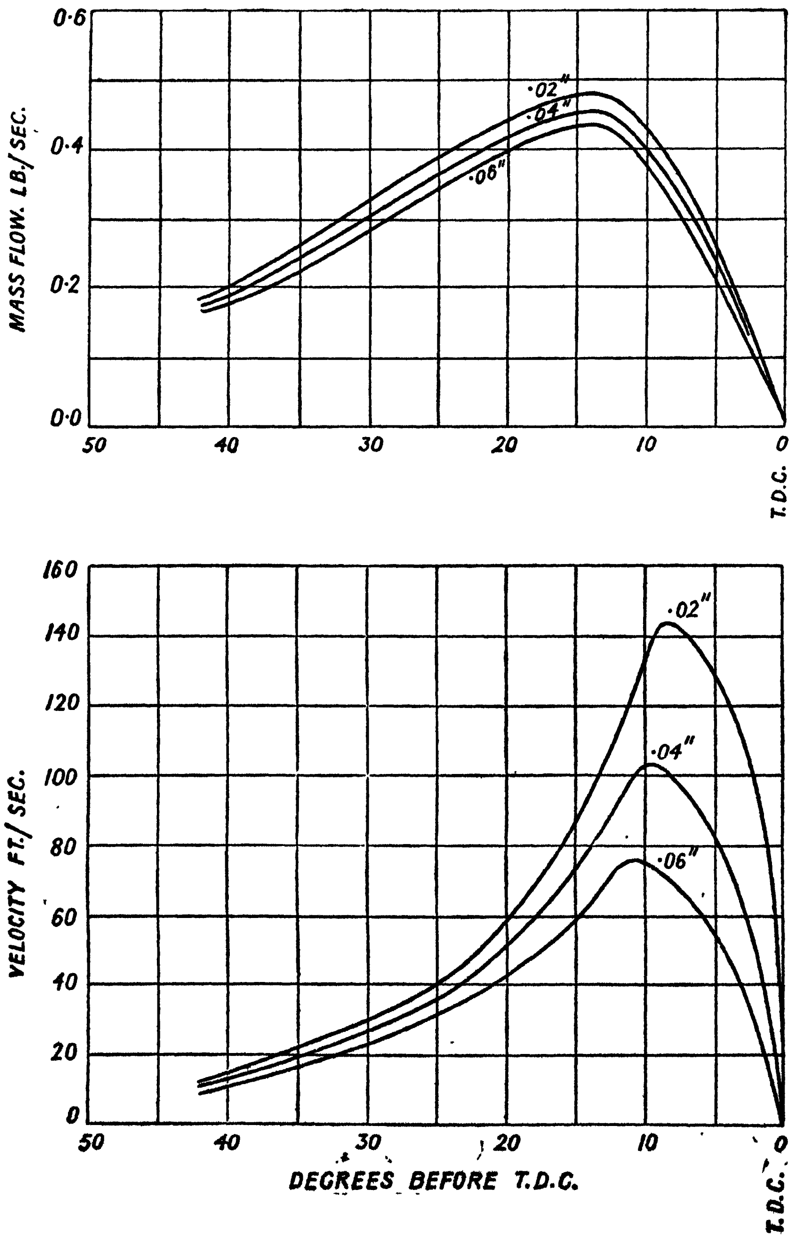


Fig. 83.—Squish effects near top dead centre

being made to other dimensions, and a change in compression ratio will result. The actual results are influenced by the way in which the change is made.

The differences in velocity indicated by fig. 83 are considerable, and represent a material difference in the energy imparted to the air during the final, and most important, stage of the compression. This energy is added immediately before the commencement of and during the first part of the injection period, and cannot fail to have a direct influence upon the combustion.

As is discussed in the following section, squish is commonly employed in conjunction with an induction-induced swirl, and to obtain an equivalent movement of the air from the induction stroke a greatly increased induction velocitv would be necessary; this would almost certainly involve some loss of volumetric efficiency and an increase in the pumping losses. The pressure differences corresponding to the transfer velocities occur during only a very small part of the compression stroke (and for the corresponding period during the working stroke), so that even if the pressures actually required are materially greater than the theoretical values, the amount of work done is small. The use of squish to augment an already existing swirl ensures that a well-defined movement is produced just at the critical moment, and it is extremely doubtful whether an equivalent effect could be produced by means of an unaided induction-induced swirl.

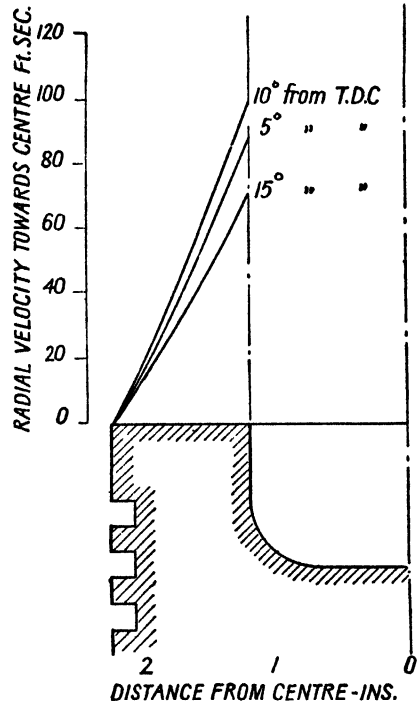


Fig. 84.—Increase in squish velocity towards the centre of the cylinder

Using the method just described, it will be possible to arrive at the velocity and the corresponding pressure at any point across the width of the annulus. Against the cylinder wall the velocity is, of course, zero, and it increases rapidly as the distance from the centre increases. Fig. 84 shows the velocity across the annulus at several points near top dead centre for the same engine as before when the

normal minimum clearance of ·04 in. is used. This diagram serves to illustrate also how rapidly the squish velocity increases as the ratio d/D decreases. In this connexion it may be mentioned that there is at least one engine in production which, using an open combustion chamber, relies solely upon squish for obtaining satisfactory combustion over a range of speeds.

The effectiveness of squish in assisting in obtaining good results from an engine may be shown in a very convincing and simple manner by rounding off the corner between the combustion chamber recess and the piston top, or slightly coning the upper face of the piston. This slight alteration, by reducing the squish velocity, may alter the air movement during the combustion to such an extent as to ruin the performance of an engine completely.

9. Combined Swirl and Squish.

In cases where a rotational swirl has been produced the effects of any squish will be added to the swirl, and a combination of the two movements has commonly been assumed to result. One effect of the squish is to provide the necessary energy for the increase in swirl velocity resulting from the transfer of the air to a smaller diameter, as has been shown earlier in this chapter.

During the transfer period, a particle of air lying close to the cylinder wall will start off with a purely rotational movement. As the compression stroke proceeds, the particle will ultimately begin to move inwards towards the combustion chamber as the air is squeezed out from the annulus. The inward velocity will tend to increase as the radial distance from the centre decreases, because the area of the path towards the centre decreases with the radius (see above). At the same time the angular velocity of the particle is increasing, theoretically as the inverse square of the radius, and the particle will move inwards along a spiral path and will finally leave the annulus with both an inward and an angular velocity, and therefore in a more or less tangential direction. The precise path followed will depend not only upon the original swirl velocity and the squish velocity, but also upon the losses sustained during the movement of the gas. The air leaving the annulus and entering the combustion chamber in a tangential direction will promote the rotational movement in the combustion recess, and as the same thing is taking place around the whole of the circumference of the combustion chamber the flow will tend to continue inwards towards the centre of the chamber. Here the meeting of the streams from around the whole circumference will cause a loss of velocity, and the resulting increase in pressure will cause a flow to take place in the only free direction, i.e. vertically into the combustion chamber.

Considered as taking place uniformly over the whole area of the

combustion chamber, this vertical velocity will be small, but if considered as taking place over only a comparatively small area near the centre, the velocity may reach a fair order of magnitude. This has commonly been assumed to be the case, and the gases have been assumed to follow a path down the centre, across the bottom of the chamber and up the sides to give a toroidal movement while they are at the same time rotating at a high velocity around a vertical axis, thus producing a movement such as a vortex ring spinning about its axis would possess.

Numerous attempts have been made to study the movement of the air during this all-important period. Alcock * is of the opinion that "... the swirling charge does not rotate as a solid mass, but rather as a free vortex, the speed of rotation increasing towards the axis", and gives as evidence the fact that a small anemometer vane gives a higher rotational speed than a large one when used to measure the swirl in the same combustion chamber. The disadvantage of anemometer tests is that the presence of the anemometer vane must introduce a serious disturbing effect to any movement other than that of rotation as a solid mass. An anemometer vane extending across the whole diameter of the chamber will, in fact, compel the air within the plane of rotation of the vane to rotate with a constant angular velocity, i.e. as a solid body.

T. F. Hurley † suggests that the air moves as a compound vortex, and as evidence quotes from some experiments carried out at the Fuel Research Station. In this instance the swirl velocity was measured by a form of Pitot tube made from very small bore tubes, which introduced only a very small disturbing element into the combustion chamber and had the advantage of allowing readings to be taken at different distances from the centre of the cylinder. By this means the velocity, or rather some sort of a mean velocity throughout the cycle, was measured at a number of different points.

The experiments were conducted upon an engine having a compression ratio of 5 : 1, but the results obtained will be similar to those obtained with a higher ratio such as is necessary for a compression-ignition engine. Describing the results, Hurley says: "The velocity decreased as the centre of the cylinder was approached, that is, an approximation to a forced vortex was obtained; but after a certain point was reached the curve departed from that of a forced vortex and tended to rise. With a forced vortex the curve would descend to the origin. Thus, after a certain point, as the centre of the cylinder was approached an approximation to a free vortex resulted. It must be remembered, however, that the curves represented an average of the velocities throughout the whole cycle and did not necessarily imply the existence of a compound vortex of the type in question right

* *Proc. I. Mech. E.*, Dec., 1934. † *Proc. I. Mech. E.*, Dec., 1934.

through the cycle. It was possible that the forced vortex predominated during the induction stroke and the free vortex during the compression and expansion stroke."

A method devised by Alcock for showing the movement of the air during the critical period consists in placing a number of dabs of thinned-out aluminium paint either on the cylinder head or piston crown close to the cylinder wall. The engine is then motored for a few minutes at the desired speed; the paint gets picked up by the rush of air during the squish period and leaves very definite markings indicative of the air movement while the piston is near its inner dead centre.

Some experiments on these lines conducted by the author show very clearly the spiral movement of the air in leaving the annulus.

Fig 85.—60° mask at 120° (max swirl), 3 35 mm gasket, 1500 r p m.

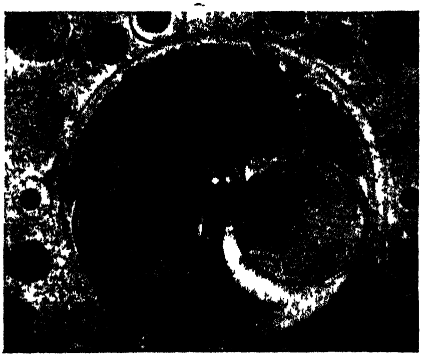


Fig. 86.—60° mask at 120° (max swirl), 3 35 mm gasket, 1500 r p m

The markings inside the combustion chamber were equally well defined; about half-way up the chamber there was a very distinct horizontal marking, indicating a layer of air having a purely rotational movement; above this line the flow lines swept upwards towards the opening and on reaching the opening pointed in the same direction rotationally as the spiral on the piston crown. Beneath the horizontal line the flow lines swept downwards towards the bottom of the chamber and passed across the bottom to the centre of the chamber, leaving a perfect spiral following *the same direction as that upon the piston crown.* In fact, when viewed directly from above, the spiral on the crown appeared to be completed by that upon the bottom of the combustion chamber. This is very clearly shown in fig. 85, which is a photograph of results obtained from one of the experiments. The marking upon the cylinder head was identical with that on the piston and showed a complete spiral as seen in fig. 86, which is the mate of fig. 85; the markings on the head show a reversed spiral as compared with the piston because it is viewed from beneath, whereas the piston is viewed from above.

Precisely similar markings were obtained from several different forms of combustion chamber: a hemispherical chamber, a truncated cone, a true toroidal form, and a cylinder with rounded bottom corners arranged to merge into a conical mound in the centre, this latter being the chamber shown in fig. 85.

Fig. 87.—No mask on valve, 1500 r p m

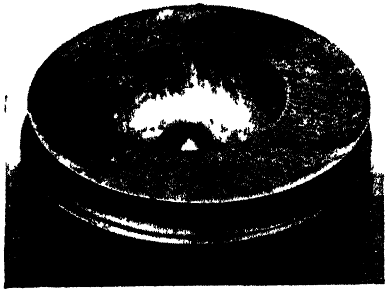


Fig 88.—60° mask at 180°, 1500 r p.m.

The results obtained from this latter chamber when swirl velocities of varying intensity were imposed upon a constant squish are shown by the series of photographs in figs. 87–92. The swirl was increased from zero (fig. 87) to a maximum (fig. 92), zero swirl being obtained by fitting a valve without a mask, while figs. 88–91 were obtained by the use of a mask extending for 60° round the periphery

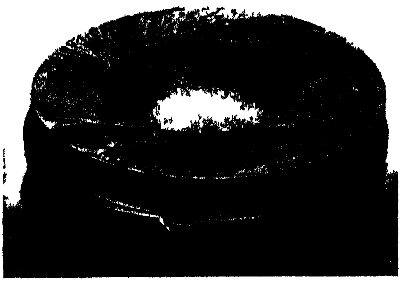


Fig. 89.—60° mask at 160°, 1500 r.p.m.

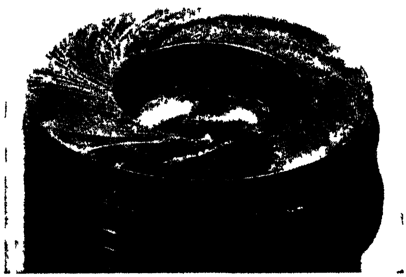


Fig. 90.—60° mask at 140°, 1500 r.p.m.

of the valve, the swirl being varied by altering the position occupied by the mask; fig. 92 was obtained using a mask occupying 120° and placed to give the maximum swirl value. In fig. 87 the markings on the upper face of the piston indicate traces of two opposing spirals. This is due to the chamber being placed slightly eccentrically in the piston top, as will be seen from fig. 85; the squish is therefore a trifle unbalanced, and there will be a flow circumferentially in both directions from the wider to the narrower part of the annulus, resulting in

a double spiral movement. Some influence will be felt from variations in the minimum clearance caused by the valves being recessed into the cylinder head. The markings within the chamber indicate a flow radially across the bottom and vertically up the sides of the chamber, indicating a truly toroidal movement such as is shown in fig. 81. This is entirely in accord with expectation, because the chamber used in these experiments, in addition to the conical central mound at the bottom, had a depth of approximately half its diameter and was therefore a form well suited to produce a toroidal movement.

The addition of only a small amount of swirl does not disturb the toroidal movement to any great extent, as is shown by fig. 88. The air is still flowing almost radially across the bottom and almost vertically up the sides, and the small amount of swirl has proved hardly sufficient to obliterate all traces of the double spiral produced by the eccentrically placed combustion chamber. In fig. 89 the spiral path taken by the air in leaving the annulus has become pronounced, while the course of the air up the sides of the chamber is at about 45° to the vertical, showing that the swirl and the squish effects inside the chamber are about equal. That the flow lines inside the chamber point in the same direction as those on the crown of the piston is clearly shown. The beginnings of a spiral movement at the centre of the bottom of the chamber were also apparent, although these do not show in the photograph. In fig. 90 the spiral marking on the top shows an increased angular component, while the flow marks up the side of the chamber are at about 30° to the horizontal, and the horizontal flow line, or " equator ", as it may conveniently be termed, can be clearly seen, although somewhat low down in the chamber. The spiral flow towards the centre on the bottom of the chamber can be seen also.

The movement shown in fig. 91 represents that obtained with the mask in the position giving excellent results from this particular engine, and represents the standard arrangement. The " equator " has risen to a point about half-way up the wall of the chamber, and the flow outwards from the equator towards the poles can be clearly seen, the direction of flow pointing more and more towards the poles as the distance from the equator increases. The direction in which these lines are pointing, however, when they reach the edge of the chamber indicates that the rotational component of the velocity is greatly in excess of that towards the pole. The direction taken by the air as it passes out of the annulus is much more nearly tangential to the circumference of the chamber than radially towards the centre, indicating that anything corresponding to the toroidal movement found in fig. 87 has entirely disappeared.

In fig. 92, which represents a condition of swirl considerably in excess of that from which the best results are obtained, the air leaving the annulus has assumed a yet more tangential direction and the

" equator " has risen somewhat higher up the sides of the chamber. The spiral movement of the air around the central mound can be clearly seen.

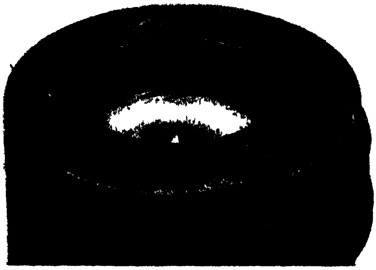


Fig. 91.—60° mask at 120° (max. swirl), 1500 r.p.m.

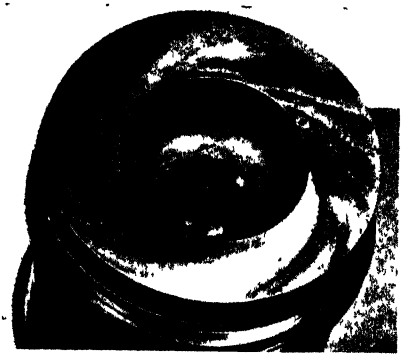


Fig. 92.—120° mask at 120° (max. swirl), 1500 r.p.m.

Fig. 93 shows how a further increase in swirl velocity raises the " equator " yet farther up the sides of the chamber and how the flow lines from the annulus assume a still more tangential direction. The course of the air winding down the chamber wall and in towards

Fig. 93.—170° mask at max. swirl, 1500 r.p.m.

Fig. 94

the centre could be clearly traced in this example. The polar component of the velocity in this case was quite small. This chamber was different from the previous ones and was in the form of a truncated cone. The combination used, however, represents the standard arrangement for an engine giving excellent results. Fig. 94 shows

precisely similar markings obtained with a hemispherical chamber.

From these experiments it appears that when squish alone is the source of the air movement the air flow in the chamber partakes of a simple toroidal movement, as illustrated in fig. 81, although to produce this movement to its full extent the chamber must be proportioned much as shown in the figure. The addition of a swirl, however, changes the nature of the movement and with a swirl of comparatively feeble proportions the toroidal movement illustrated in fig. 81 disappears completely and a movement of an entirely different nature takes its place.

It will be clear from the photographs that, under the conditions giving the best performance from these three chambers, the air after spiralling across the top of the chamber has not passed axially right down to the bottom and thence across the bottom and up the sides of the chamber to repeat the process as the upflowing stream is caught by air still issuing from the annulus, because if this represented the movement taking place *the spiral across the bottom of the chamber would be of the opposite hand to that on the crown of the piston.* The direction of rotation of the air must remain constant, so that the air moving inwards towards the centre of rotation at the top and outwards from the centre to the circumference at the bottom would describe spirals of the opposite hand, as is shown in fig. 95. Further, a movement of this kind could not account for flow lines of the nature actually found on the walls of the combustion chamber; for such a movement the flow lines produced on the walls of the chamber would be in the form of a helix running from top to bottom of the chamber, as shown in fig. 95. The markings at the two ends of the chamber indicate that precisely the same movement is taking place at each end of the chamber, namely, the air is moving inwards towards the centre and at the same time rotating around the axis of the chamber.

Any fluid rotating within a container is subject to certain well-known conditions. When it rotates with a uniform angular velocity at all points throughout its mass, as when the fluid and the container are rotated together in unison, a state of equilibrium exists. The fluid being restrained within the container, the centrifugal force produced at any point P at radius r from the centre of rotation of the mass of fluid will produce a pressure whose magnitude is proportional to the centrifugal force, and if the surface of the fluid is unrestrained, the level at a distance r from the centre will be carried above that at the centre by a distance h such that the pressure due to this head of liquid h is equal to the centrifugal pressure at the same distance from the centre. From the usual equation for centrifugal force and velocity it can be shown that $h = \omega^2 r^2/2g$, where ω is the angular velocity and g the acceleration due to gravity. This is the equation for a parabola and the free surface will assume a parabolic form. The angular velocity

being constant throughout the mass of fluid, the centrifugal pressure produced at a point P at distance r from the centre will be the same at all depths below the surface, e.g. the pressure at P distant r from the centre in plane aa (fig. 96) will be the same as at P_1 distant r from the centre in plane bb, and the whole mass of fluid will be in equilibrium, the only movement taking place being the rotational movement. This represents the conditions for a true forced vortex.

When a body of liquid is rotating inside a container, however, it is subjected to frictional resistance between the liquid and the container, and this introduces certain secondary movements which can easily be seen and studied when a glass of water, in which there is a small quantity of sediment of density similar to that of the water, is

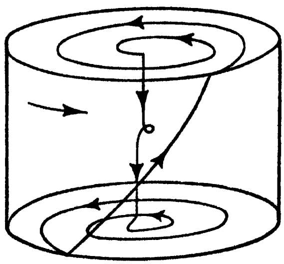


Fig. 95

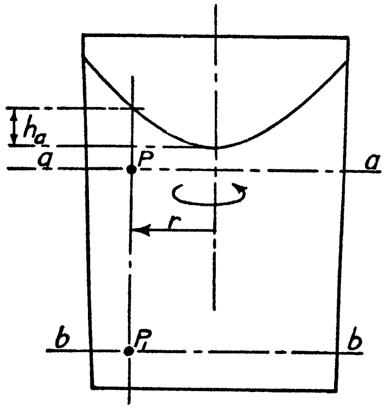


Fig. 96

stirred vigorously. The sediment will be found to travel spirally inwards across the bottom of the glass, rise upwards along the centre of rotation, and then pass spirally outwards across the surface and down the outside in a helical path, to repeat the process on reaching the bottom. With the liquid rotating inside the container the frictional resistance is not uniform at all parts; that between the liquid and the bottom of the container will be greater than that between the liquid and the air at the surface. The bottom layer of liquid will thus slow down more rapidly than the top layer, and a difference in speed will exist between the two layers. Intermediate layers will have a speed graduated from that of the top layer to that of the bottom layer, the friction between successive layers determining the speed rather than that between the liquid and the container wall, which would for equal speeds be the same for all intermediate layers.

Under these circumstances a condition of equilibrium cannot exist, as will at once be seen if we consider what is taking place at the points P and P_1 in fig. 96. If the rotational speed at plane aa in which the point

P lies is ω_a and that at plane *bb* in which the point P_1 lies is ω_b, and ω_a is greater than ω_b, then the centrifugal pressure at P will be greater than that at P_1, because the centrifugal pressure is a function of ω^2. This means that while the centrifugal pressure at P will be sufficient to support the surface of the liquid at a height h_a above the level at the centre of rotation, that at P_1 will only be sufficient to support it at some lower value h_b. The free surface of the liquid will therefore not be in equilibrium; it will be continually attempting to bring the level up to h_a by water thrown out from the centre by centrifugal force, but as soon as the water reaches the outer diameter the level will fall to h_b. There is thus a flow outwards from the centre across the surface of the liquid, down the outside and inwards across the bottom and up the centre, as is indicated in fig. 95, the flow continuing as long as the rotational movement persists.*

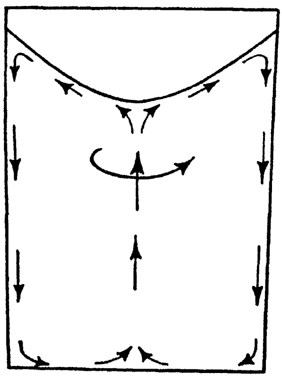


FIG 97

Viewed in cross-section this secondary flow assumes a toroidal movement, as will be seen from fig. 97. This movement will be superimposed upon the rotational movement, and the path actually taken by a particle of liquid will be as indicated in fig. 95, which, for clarity, shows the body of liquid as having a cylindrical form.

The movement of air rotating within the combustion chamber will be subject to precisely the same influences as that of the water in the glass. There is, however, an important difference between the two; whereas the water has one free surface, that between it and the air, at which the friction loss is a minimum, and one surface in contact with the end of the container, at which the friction loss is a maximum, the air is in contact with the container at both ends of the combustion chamber and therefore has a point of maximum resistance at each end. The difference does not introduce any difference in principle, although it does modify conditions and the actual movement of the air quite considerably. The effect of two closed ends is to increase the friction at the two ends of the rotating mass of air and bring the layer subject to the minimum frictional effect, i.e. the layer having the maximum velocity, to a point somewhere between the two ends, although not necessarily midway between them, because the velocities at the two ends are not necessarily equal. The plane in which the

* Fig. 95, having been drawn to conform to an inward movement at the top, such as actually occurs in the engine, must be looked at upside down to give the movement in a glass of water.

maximum centrifugal pressure is produced is thus somewhere near the middle of the length of the rotating mass, and a flow will take place from this plane towards both ends, thence across the two ends inwards to the centre, and then along the axis. The two streams flowing towards each other along the axis will meet at, or about, the plane of maximum rotation speed and then flow outwards towards the circumference, the whole movement being as represented in fig. 98. As will be seen, this figure is fig. 95 duplicated, with a plane of symmetry at the plane of maximum rotational velocity. Viewed in cross-section the secondary movement assumes the form of a double toroidal movement, as indicated in fig. 99. Owing to the fact that the air is moving inwards towards the centre at both ends of the chamber, the air in crossing the two ends will describe spirals of the same hand. A spiral of opposite hand to the two ends will be described at the plane of maximum rotation, as at this plane the air will be moving from the centre outwards.

Such a movement entirely agrees with the markings obtained during the experiments just described and illustrated in figs. 87–92. The fact that spirals of the same hand were obtained at each end is satisfactorily accounted for; the "equator" represents the plane of maximum revolution from which the air flows endways towards the poles, with an increasing polar velocity component as the ends are approached and the additional frictional resistance of the ends makes itself felt more and more. The spiral of opposite hand as the air moves outwards again from the centre naturally leaves no trace, since it takes place in space in a plane somewhere between the two ends of the chamber.

The evidence obtained from figs. 87–92 proves that when a rotational velocity of any magnitude is present, the imposition of a squish movement upon this rotational movement does not result in a movement such as is indicated in fig. 95. In figs. 87–92 the squish was produced by a minimum clearance between the piston and cylinder head of 0·064 in. Fig. 100 shows the effect of reducing the minimum clearance to 0·032 in., while the mask was in the same position as when fig. 91 was obtained. This example, instead of showing any tendency towards producing the movement indicated in fig. 95, indicates that the rotational movement has been increased. An examination of figs. 87–92 shows that the purely toroidal movement obtained when squish alone is provided (fig. 87) is very quickly suppressed when a swirl is provided also. Fig. 89 shows that when the swirl and squish velocities are about equal in magnitude, as shown by the angle made by the flow lines with the plane of the piston top, the simple toroidal movement has almost disappeared, while fig. 90 shows that when the rotational velocity has reached a magnitude of about twice the axial, or polar, velocity all trace of the toroidal movement shown in fig. 95 has

vanished and the double toroidal movement shown in figs. 98 and 99 has appeared. This double toroidal movement, however, is not the product of the squish; it is the natural movement resulting from the rotation of a fluid in an enclosed container and, as fig. 100 shows, the

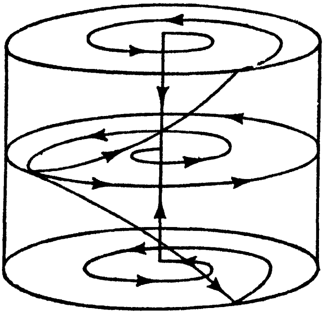


FIG 98

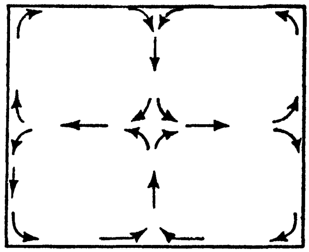


FIG. 99

application of a more powerful squish results not in a more marked double toroidal movement, but in a relatively less marked one. At first sight, a really marked increase in squish, as shown in fig. 100, might be expected to promote the movement indicated in fig. 95, rather than to enhance a double toroidal movement, but in actual fact, it does neither of these things, but serves only to augment the swirl.

The effect of the squish is therefore not to impose a secondary air movement on to a swirl which has previously been produced, but to increase the swirl already present. It provides the energy necessary to transfer the air from the larger diameter of the cylinder bore to the smaller diameter of the combustion chamber, and in doing so to accelerate the swirl in accordance with the principle of conservation of moment of momentum. This is the sole effect of the squish. The double toroidal movement indicated in the swirl experiments described earlier is produced by frictional effects and is only connected with the squish in so far as an increase in velocity results in an increase in frictional effect. The single toroidal effect combined with a rotational movement, such as is illustrated in fig. 95, exists only when the swirl

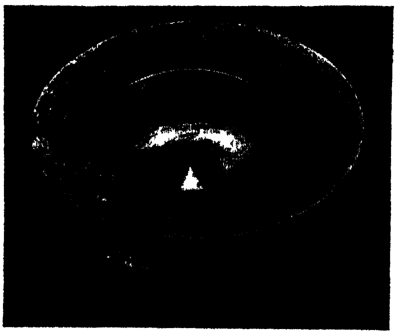


FIG. 100.—Swirl marking at 1500 r.p.m. when clearance between piston and head is reduced to ·032 in.

velocity is of a very low order, much lower than that necessary to give the optimum results in the engine. Under these circumstances it might be argued that the squish really serves no useful purpose, since a swirl of sufficient magnitude can be produced by a suitable mask on the inlet valve. This may be true, but the presence of a powerful squish serves two very useful purposes: (1) it enables an induction-induced swirl of much lesser magnitude to be employed, i.e. a much smaller mask can be used, with a corresponding reduction in air flow resistance, and (2) it builds up the swirl just when it is most wanted, i.e. immediately before the injection period. As an example of the reduction in mask size which results from an adequate squish, two engines built by the author's firm may be quoted. One which has an individual cylinder capacity of 1475 c.c. needs a mask occupying only 60° of the circumference of the valve, and the other, with a cylinder capacity of 1650 c.c., needs a mask of only 45°, to produce the requisite amount of swirl.

10. Compression-induced Swirl.

Swirl which is generated during compression is obtained by transferring the charge of air during the compression stroke, almost wholly or in part, from the cylinder to a separate chamber. The shape of this chamber, and the connexion thereto, are arranged in such a way that the air is given a rotary movement. This chamber forms the combustion chamber, and into it the fuel is introduced, and as expansion takes place the contents of the chamber are retransferred to the working cylinder. Usually the swirling movement of the air is confined to this separate chamber, although in certain cases some degree of swirl is given to the air in the space above the piston.

For mechanical reasons it is not possible to transfer the whole of the air to this separate chamber. Some clearance is necessary between the piston when at its inner dead centre and the cylinder head, and although this clearance is kept down to quite small dimensions the volume of air retained in it will usually represent about 20 per cent of the total clearance volume. In certain designs a definite fraction of the compressed charge is retained outside the separate chamber and is then accommodated in a suitable cavity in the piston crown.

The compression of the fresh charge into a chamber separate from the main cylinder volume dates from the earliest days of the oil engine. The original Akroyd-Stuart engine was provided with such a chamber, but the shape of the chamber and the connexion between it and the cylinder were not such as to produce a well-ordered swirling movement (although there may, perhaps, have been some tendency towards a toroidal movement), but were calculated rather to produce a turbulence of an indiscriminate nature. Many of the hot-bulb two-stroke

engines had a separate chamber, but here again the air movement was more of a turbulent nature.

The swirling movement may be of two kinds, a purely rotary movement where the mass of air, either in a cylindrical or a spherical shape, is made to rotate about an axis, or a toroidal movement, where the air is made to rotate in a manner similar to that of the familiar smoke ring.

The chambers are variously placed, either central with the cylinder, in which case they are usually accommodated in the cylinder head, or to one side of the cylinder bore, when they may be carried either in the cylinder head or in the upper part of the cylinder casting. Chambers giving a toroidal movement are naturally placed centrally over the cylinder, because of the necessity of obtaining a uniform movement throughout the torus of air. Any lack of symmetry in the movement would cause eddying and a rapid loss of motion and at the same time would result in an uneven distribution of the air, to the obvious disadvantage of the engine.

The size and shape of the aperture to the chamber will have a very direct bearing on the rate of air swirl which is obtained. With the necessarily small volume of the chamber there can be very little latitude as to its dimensions, so that the swirl value cannot be varied very much by a change in the size of the chamber itself. Where a cylindrical form is chosen there is a certain amount of choice as to the length-diameter ratio, but even here the necessity for providing a reasonable distance for spray travel imposes quite definite limits. The designer, therefore, relies mainly upon the aperture for obtaining the desired rate of swirl, and fortunately the aperture provides ample means for obtaining all the swirl that may be required.

The connecting passage may be made either clear and unobstructed, i.e. with the same area throughout the cycle, or may be so arranged that at the inner end of the compression stroke the area of the opening is reduced, either by the piston itself or by a protuberance provided on the piston for this purpose. The idea of this latter arrangement is to have a comparatively large opening which offers a relatively low resistance for most of the compression stroke. With such an arrangement the velocity through the passage during the early stages of compression will be low and will generate only a low rate of swirl, but during the latter stages the area of the opening is reduced by the piston, so that the velocity through the opening is increased and thus increases the rate of swirl. With an unobstructed passage the rate of flow will be greater during the earlier stages of compression, but if the same final rate of swirl is to be produced with both types, the velocity through the passage when an obstructed passage is used must be made greater during the later stages in order to make up for work which has not been done during the earlier part of the compression

stroke. As the density of the gases is greatest during the final stages of compression, the work done during this period must be substantially increased and thus counter-balances any saving made during the earlier stages. The only real justification for the obstructed passage is when the valves are arranged to open into the combustion chamber instead of opening directly into the cylinder. In this case an orifice of the size necessary to pass the gases during the suction and exhaust strokes would be totally inadequate to produce a swirl of sufficient magnitude to serve any useful purpose. It is therefore necessary to reduce the opening during the final stages of compression in order to boost up the velocity and so produce the requisite amount of swirl.

11. Combustion-induced Swirl.

Combustion-induced swirl is obtained by making use of the sudden displacement of the air which follows naturally upon the ignition of the fuel. The sudden rise in temperature occasioned by the ignition of the fuel causes a flow of gas away from the point of ignition, so that a definite movement of the air can be produced by ensuring that ignition will always take place in a predetermined zone of the combustion chamber and by suitably shaping the chamber itself.

To secure the desired effect it is necessary to separate the air into two chambers, one usually being materially larger than the other; it is in the smaller of the two that ignition takes place. The ignition of the fuel causes a sudden increase in pressure within the smaller chamber and forces the bulk of its contents out into the larger chamber. The rush of gases and fuel from the smaller chamber sets up a movement of the air within the larger chamber, this latter being so shaped that the desired type of movement is procured.

This form of combustion chamber has points of similarity to both the air cell chamber and the pre-combustion chamber, but whereas in the pre-combustion chamber the whole of the fuel is introduced directly into the smaller chamber, only a part of the fuel may be introduced into the small chamber of the combustion-induced swirl chamber and is arranged to reach the small one via the larger instead of being delivered directly into the small chamber. Perhaps it might be more correct to describe this type of chamber as an air cell chamber in which provision is made to ensure that the contents of the main chamber are given a definite swirling movement by the disturbance produced by ignition.

The intensity of the swirling movement will depend upon the form of the main chamber and the rate at which the contents of the small chamber are discharged into the main chamber. This will in turn depend upon the volume of air contained in the small chamber and the quantity of fuel involved in the initial combustion. There is thus

no definite relationship between the speed of the engine and the rate of swirl. Some sort of relationship will be obtained from the increase in ignition delay as the speed increases, because an increase in delay may result in more fuel entering the chamber and hence in a greater pressure rise, and a greater subsequent rate of discharge, provided of course that the cell contains a sufficient quantity of air to burn this added quantity of fuel. It thus appears that where a combustion-induced swirl is used the engine will not be capable of so wide a speed range as with either induction-induced or compression-induced swirl. By suitable proportioning, however, excellent results can be obtained at any given speed and for a moderate range of speeds on either side.

12. The Persistence of Swirl.

The orderly swirling movement of the gas within the cylinder has an astonishing amount of persistence. One hears it said that the disturbance caused by combustion will effectively destroy the swirl. This most certainly is not the case, as the following proves. During some early experiments with an open-type combustion chamber it was found impossible to obtain reliable exhaust gas samples for analysis. With a swirl chamber engine and an air cell engine the samples had always been consistent and reliable, and the results obtained from them agreed very well with those given by the actual measurement of the air and the fuel. It was found also that with the open chamber the sample varied with the engine speed, quite a small change in speed giving a marked change in composition of the exhaust. Further, a change in the amount of swirl produced a large change in the composition. Both of these changes might with some justification be argued as due to changes in combustion conditions brought about by the change in engine speed or swirl speed, and it was of some importance to know whether this was the case. The fact that a change in speed which did not result in a change in specific engine performance produced a material change in exhaust composition went to show that a change in combustion efficiency was not the cause of the change in exhaust composition, and an investigation into the question was therefore instituted.

The engine had a single cylinder and the sample had been collected by a small pipe brought close down on to the head of the exhaust valve, the object of this arrangement being to avoid the possibility of error due to air being drawn into the exhaust pipe through a worn valve guide or faulty pipe joint by the momentum of the column of exhaust gas. This arrangement had previously been successful in giving accurate samples, and was therefore the first arrangement employed with the open chamber. After making the valve guide a good close fit and making sure that the joint between the pipe and the head was in good condition, a sampling pipe was fitted in such a way that it

could be swung across the diameter of the port, and a sample was taken at points a small distance apart across the whole diameter of the port. The process was repeated across a diameter at right angles to the first. It was found that the composition of the gases varied greatly across the pipe, and not only this, but the distribution changed with a change in both engine speed and swirl speed. It was noted also that when the engine had been running for some time under a fixed set of conditions the soot in the pipe showed a helical marking. This gave the clue. The helical marking might, of course, have been produced by a rotary movement in the exhaust gases caused by the shape of the port, but it might also be due to a residual movement from the movement within the cylinder.

A straight-vaned anemometer was therefore mounted in a short stub pipe fitted to the port, and although the anemometer could not be made to work satisfactorily with the engine running, owing to the jamming action of the end thrust produced by the rush of the gases, a very pronounced swirl was produced when the engine was motored. The speed of the swirl in the exhaust pipe varied with the swirl in the cylinder, and, most conclusively of all, the swirl in the exhaust pipe reversed when the direction of the swirl in the cylinder was reversed by bringing the mask of the inlet valve on to the other side of the cylinder. At an engine speed of 1500 r.p.m. a swirl speed in the exhaust pipe as high as 5000 r.p.m. was recorded.

This experiment showed clearly that the swirl, once produced, has a great degree of persistence, even to the extent of continuing in the exhaust pipe. It appears that the rotating cylinder of air maintains its form throughout the combustion period and expansion stroke, merely expanding as the piston moves outwards, without indiscriminate mixing to any great extent. Any indiscriminate mixing would result in a fairly uniform composition of the exhaust gases, whereas the composition was found to vary steadily across the pipe, the maximum concentration of carbon dioxide being at the centre of the pipe.

This experiment goes far to explain why in an open-type combustion chamber the injection of an excess of fuel into part of the air is not rectified later in the cycle, and why a vigorous swirl is necessary if a high specific output is required from a high-speed engine. The fuel, once distributed in a certain way across the combustion chamber, retains that distribution more or less unaltered throughout the expansion stroke and even to some extent out into the exhaust pipe, although here a certain amount of mixing will almost certainly have taken place. The condition is similar to a case of segregation or piping in an ingot of steel, the fault remaining across the cross-section even after the ingot is drawn down into a small wire.

It may be argued that the indiscriminate turbulence produced during induction will have the effect of distributing the fuel and air

more or less evenly, although perhaps too late to be of any assistance in making the best use of the fuel. The answer to this is that the indiscriminate turbulence, consisting as it does of whirls and eddies, results in a considerable amount of friction. This friction is greatly increased by the high compression and the compact form of the compressed mass of gas; the eddies therefore die out very quickly and have mostly ceased to exist by the end of compression. In any case, the admission of the air in a definite direction tends to minimize this indiscriminate turbulence. The swirl, on the other hand, results in the whole mass of gas rotating more or less as a solid body, and although there is friction against the cylinder walls which will produce a certain amount of damping, the dynamic energy in the gases is sufficient to ensure a continuation of the rotation for an appreciable period of time, certainly for a greater length of time than that necessary to cover the essential part of the cycle.*

13. Swirl Speed and Engine Speed.

In order to accommodate varying engine speeds the swirl speed should vary directly with the engine speed. Theoretically, for a given fuel delivery the injection period of a "jerk pump" system remains constant, and the rate of fuel injection therefore varies directly with the engine speed. To bring the fuel and air together at the desired rate the movement of the air must be increased at the same rate as the delivery of the fuel, i.e. it must vary directly with the engine speed.

The statement that the rate of fuel injection varies directly with the engine speed requires a certain amount of qualification, because in actual fact some variation from this desirable condition is apt to take place, especially when nozzles which offer a considerable amount of resistance are used. The compressibility of the fuel and spring in the pump mechanism can lead to quite an appreciable departure from strict proportionality, as is discussed later when dealing with injection systems.

A swirl movement which is dependent upon the displacement of air by the movement of the piston will obviously tend to keep step with the engine speed and will respond directly and instantly to changes in speed, although the elasticity of the air introduces a disturbing factor tending to cause a departure from true proportionality. Induction-induced swirl and compression-induced swirl are well adapted to enable an engine to operate over a wide range of speeds, and it can be said that generally speaking the air movement is not a factor limit-

* Since the above was written, confirmation of these conclusions has been given in Report No. 650 of the National Advisory Committee of Aeronautics, Washington (*A Study of Air Flow in an Engine Cylinder*, by D. W. Lee). In this it is shown that indiscriminate turbulence dies out very quickly, but "any horizontal rotation set up during the intake stroke always persists, at a decreasing rate, throughout the compression, the expansion, and the exhaust stroke".

ing the engine speed, although with certain types of nozzle there may be some tendency for the air and fuel to get out of step at extreme speeds.

Where the air movement is dependent upon a combustion-induced swirl, however, conditions are somewhat different. The rate of swirl is dependent upon the initial rate of combustion, which generally speaking bears little relationship to engine speed, and this system is therefore not so well adapted to give a wide range of engine speeds. Some compensation will be obtained from an increase in delay period as the speed increases. An increased delay means a greater quantity of fuel present at the moment of ignition and therefore a more violent initial combustion, with a higher maximum pressure and a correspondingly more vigorous swirl.

The selection of the method of swirl production will be largely dependent upon the service for which the engine is intended. For a wide speed range at maximum torque induction- and compression-induced swirls are most suitable, whereas for a constant-speed engine, given a suitable design of chamber, all three have points in their favour, although here again the general preference is for induction- or compression-induced swirl. Where an exceptionally wide speed range at maximum torque is desired, compression-induced swirl would appear to be most suitable, because (1) it does not necessitate a type of nozzle which tends to produce a very marked change in injection rate as the speed changes, and (2) it does not in any way interfere with the breathing of the engine. On the other hand, the work done in producing the swirl tends to increase with speed and has a somewhat adverse effect upon the mechanical efficiency of the engine. The combustion efficiency is, however, unimpaired, and although the specific fuel consumption is not so good as at lower speeds, a clean exhaust is maintained over the whole speed range.

14. Swirl and Rate of Combustion.

In addition to ensuring the correct distribution of the fuel and the air, the swirl performs another important function, that of maintaining the correct rate of combustion when a change in engine speed has taken place. The air flowing across the path of the blazing fuel particles promotes rapid combustion just as the air flowing through a furnace does, and as in the latter, the rate of combustion is directly connected with the rate of air flow. The rate of burning of a fuel particle is proportional to its velocity relative to the air. A change in engine speed, by changing the rate of swirl, increases this relative velocity and thus produces a change in the rate of combustion proportional to the change in swirl, and where the swirl velocity is proportional to the engine speed the rate of combustion is maintained proportional to the engine speed.

This is very important from the point of view of engine efficiency, because if the efficiency is not to be affected by a change in engine speed the degree of completeness of combustion attained at any given point in the cycle must remain constant regardless of the speed of the engine. If there is any tendency for the engine speed to outstrip the combustion, it means that burning continues farther down the expansion stroke and the mean expansion ratio will be reduced with a corresponding reduction in efficiency. If, on the other hand, the rate of combustion is increased relative to the engine speed, which very commonly happens at the lower speeds, then the maximum pressures will be increased, and although the efficiency of the cycle may perhaps be increased thereby, the increased stresses and loading on the bearings and also a tendency towards roughness have to be taken into account. The aim of the designers should be to determine the best all-round rate of burning as measured in terms of the individual cycle, and then to reproduce it at all speeds at which the engine is intended to operate.

The fact that an increased rate of pressure rise or an increased maximum pressure follows upon a change in speed does not necessarily mean that the true rate of burning relative to the engine speed has increased. The influence of the delay period has to be taken into consideration also, because an increase in delay will produce an increase in the rate of pressure rise and in the maximum pressure. It is not only at the lower speeds that an increased rate of burning occurs; the author has known an instance in a swirl chamber engine where the rate of burning showed an increase as the speed increased. This was associated with a somewhat excessive rate of swirl and a marked increase in compression temperature as the speed increased. The delay period remained practically constant over a wide range, but the rate of pressure rise and the maximum pressure both showed a marked increase as the engine speed increased. A reduction in swirl rate resulted in more nearly constant combustion conditions.

15. Swirl and Delay Period.

Swirl has little, if any, effect upon the duration of the delay period, and in point of fact the evidence from bomb experiments seems to point to such small effect as is produced being adverse rather than advantageous. Neumann * states that whereas in still air he obtained ignition in his bomb with a temperature of 265° C., with a fan in operation to produce turbulence the temperature had to be increased to 306° C. before ignition took place. Up to a temperature of 390° C. the delay was increased by the fan, but above this temperature the delay was decreased. Bird,† however, states that by the use of such fans, "... it was possible with small jets to suppress ignition com-

* *Z. V. D. I.*, Vol. 70, p. 1071 (1926). † *Proc. Inst. Mech. E.*, 1927, p. 1030.

pletely (a phenomenon also possible to produce in an engine). No evidence of a reduced time lag was obtained over a range of speeds."

The compression temperature normally found in high-speed engines is considerably higher than the figure of 390° C. referred to by Neumann, and under engine operating conditions there does not appear to be any evidence of any marked influence of swirl upon delay. Some indirect effect upon delay is, however, obtained by the effect which swirl may have upon compression temperature. Swirl may result in either an increase or a decrease in compression temperature, and will therefore have a corresponding influence upon delay. While this is caused by the swirl, it is not strictly an effect of swirl as such, and it appears that, other things being equal, swirl does not influence delay to an extent that is reflected in engine behaviour.

CHAPTER VIII

Types of Combustion Chamber

1. Types of Combustion Chamber.

Innumerable different forms of combustion chamber have been used for high-speed compression-ignition engines; it is not, however, the author's purpose to attempt to catalogue all these various designs, but to discuss the principles underlying the main types and to give representative members of these types.

The different types of chamber may be divided into four classes: open combustion chambers, pre-combustion chambers, swirl chambers, air cell chambers.

The object of all these chambers is the same, namely, to bring the fuel and the air together in such a way that a satisfactory performance may be obtained from the engine. Reduced to rock bottom, the multiplicity of designs of combustion chamber is due to the shortcomings of the fuel-injection apparatus, which necessitate the adoption of some means for doing work that the injection apparatus has failed to do. Having failed to make the fuel woo the air with sufficient ardour to attain the desired end, we are constrained to make the air take a hand in the wooing and meet the fuel half-way at least.

The qualities required from the engine are as follows: economy, clarity of exhaust, smoothness of operation, flexibility, reliability, simplicity, ability to use a wide range of fuels.

Some of these qualities are conflicting and the encouragement of one sometimes leads to the deterioration of another, and it is therefore the function of the combustion chamber to blend together all the desired qualities in such a way that the resulting compromise will be a sound commercial proposition. The stressing of one quality to the exclusion of the others will never produce the desired result.

The actual order of importance of the various qualities will depend somewhat upon the service for which the engine is intended, so that the relative importance of each may change accordingly. It is not possible, therefore, to be dogmatic as to the relative merits of any one quality, but a few words in amplification of each item in the list will not be amiss.

Economy scarcely needs to be enlarged upon. It is the high fuel

economy of the compression-ignition engine that forms its chief attraction, and is the quality it can claim to possess beyond any other form of heat engine. The economy of the compression-ignition engine is almost entirely free from size effects; the fuel consumption of the smallest units can be made almost to equal that of the best of the largest, and in many instances the small engine shows a better specific consumption than some of its larger brethren. Unfortunately, economy is one of the qualities that is apt to run counter to some of the others, and in straining for the absolute maximum of economy, smoothness and reliability may suffer. The maximum economy will be obtained by burning the whole of the fuel at constant volume, and the attainment of a high percentage of constant volume burning is associated with high rates of pressure rise and also high maximum pressures. Both of these features tend to have an adverse effect upon reliability, and in addition to this a high rate of pressure rise causes rough and noisy running. In the interests of smoothness and reliability, therefore, it is frequently necessary to be satisfied with something less than the maximum economy of which the engine is capable. Taken in its strictest sense, economy includes reliability also, because in a commercial sense economy means the total cost per unit of useful work and includes not only the cost of the fuel but of everything necessary for keeping the engine running, as well as the original cost of the engine, and also depreciation. Thus an increase in commercial economy may sometimes be obtained by the sacrifice of a fraction of thermodynamic efficiency—a condition which from the purely engineering standpoint must be deplored.

Flexibility and fuel economy are inclined to run counter to one another. In certain services flexibility means not only the ability to obtain wide variations in load at one given speed, but also the ability to vary both the load and the speed simultaneously over a wide range and with frequently recurring changes. This latter quality is not always required, many engines being required to vary only the load they carry without any change in speed, a condition which is fairly easily fulfilled without any conflict with economy. Other engines may be required to develop their maximum torque at one fixed speed, and if the torque is reduced then the speed decreases also. This again is fairly easy of attainment without conflict with economy, because the main operating range of such engines is between fairly narrow limits near the designed maximum, and the periods of operation outside these limits are usually so short that if the economy outside the normal range is not all that can be desired, the effect upon the operating costs is too small to be of serious account.

Engines such as are used for land transport must operate at widely differing, and rapidly changing, loads and speeds. A speed range as wide as possible must be obtained, and the economy must be main-

tained at a high figure over the whole range of both loads and speeds, a condition by no means easy of fulfilment. The exhaust, too, must retain its clarity over the whole range of loads and speeds; a degree of discoloration which would cause no comment for other services cannot be tolerated in a vehicle engine, and although a clear exhaust will never conflict with economy, a clear exhaust and flexibility do not always go hand in hand.

Simplicity does not clash with any other quality, but in many cases a loss of simplicity results from the attempt to bring the other conflicting elements into some sort of harmony, and a loss of simplicity is apt to react unfavourably upon both first cost and reliability.

The ability to use with equal facility fuels of widely differing characteristics is a great asset in an engine. In this country, where fuels of high ignition quality are available everywhere, there is, perhaps, a tendency to lose sight of this point, but in the case of engines which are intended for export to all parts of the world the ability to use a wide range of fuels is a matter of first importance, because in a great many parts of the world fuels of high ignition quality are not readily obtainable. In localities where the demand for fuel oils of any kind is relatively small the range of fuels available is usually narrow, and each fuel is made to embrace as wide a scope as possible. For obvious reasons the supplies are drawn from the nearest convenient oil field, and it may happen that the crude from this field is not one from which a fuel of high ignition quality can be produced. In point of fact, the number of fields producing crudes from which fuels of high ignition quality can be distilled are relatively few, and in the more out-of-the-way parts of the world the fuels available are therefore more likely to be of low, or relatively low, ignition quality rather than the reverse. Furthermore, the fewness of the areas producing fuels of high ignition quality gives food for thought as to the length of time these high-quality fuels will continue to be available even in this country. It would seem that, with the tremendously rapid increase in demand for such fuels that has taken place in recent years, the time must come, and at no very distant date, when, if the demand goes on increasing, some deterioration in ignition quality will take place. Under these conditions the ability of an engine to use satisfactorily fuels of a relatively low ignition quality will become a matter of increased importance, and is therefore a point which merits careful consideration.

Conditions on the North American continent afford an interesting example of the need for engines capable of using a wide range of fuels. Fuels are produced in large quantities in different parts of the continent from crudes of widely differing characteristics. The fuel available at any particular spot is governed by the nearest convenient source of supply, so that the nature of the fuel varies in different parts of the

continent. Within certain limits, engines for stationary purposes can be, and usually are, adjusted to suit the type of fuel available in their neighbourhood, but vehicle engines and, to a lesser extent, engines used in railway service, travelling as many of them do all over the continent, are called upon to utilize whatever fuel is available in the neighbourhood in which they happen to be when replenishing their supplies. The wider the range of fuels that the engine can utilize without trouble, or adjustment, the wider the market it will command.

2. The Open Combustion Chamber.

The open combustion chamber—or, as it is also called, the direct injection chamber—is the simplest and most straightforward form of chamber. As its name implies, the chamber is really a part of, or a direct extension to, the engine cylinder itself, although it rarely, if ever, conforms strictly to this definition. In the interests of compactness the diameter of the chamber is made somewhat smaller, often considerably smaller, than the diameter of the cylinder. The chamber may be either situated in the cylinder head or recessed into the crown of the piston or partly in both. In by far the largest number of designs, the necessity for accommodating the valves in the cylinder head makes it difficult, if not impossible, to accommodate the chamber in the cylinder head, and it is therefore placed in the piston, where it is situated centrally or very nearly so. An organized air movement being essential for high-speed operation, the chamber is invariably circular in plan, but in cross-section numerous different shapes are employed.

The open combustion chamber has the advantage that, apart from the act of compression which is common to all engines, very little work is done upon the air. The air movement necessary in order to obtain rapid and complete combustion is obtained primarily during the induction stroke, where small pressure differences are adequate to produce a high velocity, and, although in an extreme case a serious loss of volumetric efficiency might be caused, the actual amount of work expended in producing the air movement is very small. The pressures induced during any squish period may reach fairly high values, but as they are confined to a very small movement of the piston towards the end of the stroke, no very great amount of work is done and the mechanical losses of the engine are not seriously increased. As a result, the mechanical efficiency of engines fitted with an open combustion chamber is usually high, and the fuel consumption and the efficiency on the B.H.P. are of a high order and are, or should be, better than those obtained with other forms of chamber. This high efficiency is a very material advantage and goes far to outweigh certain disadvantages.

On the other side of the scale, it is difficult to obtain a satisfactory distribution of the fuel and the air at all conditions of load and speed,

and as a result the open chamber is much more prone to have a discoloured exhaust than are some other types of chamber, and for the same reason it is more difficult to maintain a high efficiency over a wide range of speeds. The degree of air utilization is usually of a relatively low order, about 60 per cent of the oxygen, and even less in many cases, being as much as many engines of this type are capable of utilizing, although the loss of useful power, which this might otherwise entail, is sometimes made up by the high mechanical and thermal efficiencies. The air utilization may be considerably improved by the provision of a powerful squish, which also has the effect of improving the speed range over which the maximum efficiency is maintained, and of giving an improved result throughout the whole speed range of the engine.

The multi-hole nozzle commonly used with engines of this type is more subject to derangement by carbon formation than are some other forms of nozzle, and any carbon formation much more quickly results in a smoky exhaust. The form taken by most chambers does not lend itself easily to the provision of hot surfaces from which the fresh charge may extract heat and so raise the compression temperature, and, as a result, the compression temperature under normal working conditions is usually somewhat lower for a given compression ratio than with some other forms of chamber, and there is thus a tendency towards a somewhat longer delay period and also to a greater increase in the delay as the speed increases. This factor, coupled with the greater proportion of the finer particles of fuel associated with the multi-hole sprayer, causes a certain harshness of running. This harshness is accentuated by the fact that the initial rise in pressure takes place in direct contact with the piston, and there is therefore nothing to moderate the shock as is the case when ignition takes place in a separate chamber. Certain examples of the open chamber which use a single orifice of relatively large diameter are notable exceptions to this harshness. The comparatively coarse spray produced by the large single hole burns more slowly, and so moderates the initial rate of pressure rise and gives a remarkable freedom from harshness.

The imposition in this country of heavy taxation on fuels for road transport purposes has given an added interest to the high economy possible with this type of chamber, and much work has been done upon it during recent years, and is still going on, to overcome the drawbacks and improve its performance in other directions. Careful study of air movements and fuel injection problems has resulted in improvements in both the air utilization and the speed range over which good combustion conditions can be obtained, and the type is increasing in popularity.

3. Forms of the Open Combustion Chamber.

The simplest forms of this chamber are the hemisphere—or a segment of a sphere which is not far removed from a hemisphere—and the cylindrical, both of these forms being used in highly successful designs. With a hemispherical chamber (fig. 101), an outstanding example of which is found in the Gardner engine, a vertical, centrally disposed multi-hole sprayer is used, the number of holes varying from two upwards, although four is perhaps the most usual number. An air swirl is provided by means of a masked inlet valve, although instances can be found where a tangentially arranged inlet port is used and no mask provided on the inlet valve. The velocity of swirl necessary will be governed by the number of sprays from the nozzle, while the actual value of the swirl velocity will depend upon the arrangement of the mask on the valve and the amount of squish given by the piston. A hemispherical chamber does not give very much latitude as to the amount of squish which can be obtained, because, with a given compression ratio and cylinder capacity, the size of the hemisphere is automatically fixed, so that, unless a small stroke - bore ratio is used, the diameter of the hemisphere usually represents a fair proportion of the cylinder diameter. The diameter of the hemisphere may be reduced somewhat by adding to it a short cylindrical extension, but even with this expedient the squish effect is restricted as compared with that which it is possible to obtain with some other forms of chamber. The minimum clearance possible between the piston and the cylinder head at top dead centre will be the same for all shapes of chamber, and it is not therefore possible to gain much by a reduction in this dimension.

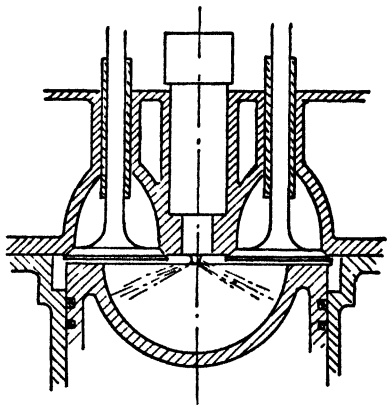


Fig. 101.—Open combustion chamber: hemispherical combustion recess

By making the chamber a segment of a sphere the squish effect may be eliminated entirely if so desired, although, when high speeds are required, this is hardly likely because it is at the higher speeds that the squish is found to be most beneficial in obtaining a high efficiency and a clear exhaust. At high speeds a high rate of swirl becomes necessary, and without a squish the masking of the inlet valve would have to be considerable to give the required swirl speed, and some loss of volumetric efficiency would certainly result.

With chambers approximating to the hemispherical, the holes in the nozzle are arranged so that they will lie around a cone having an apical angle of about 120° to 130° or a trifle more, angles around these figures giving the best distribution of fuel throughout the hemisphere of air. With engines of the larger sizes an additional hole directing a jet of fuel vertically downwards is sometimes provided. There is, however, no hard-and-fast arrangement, and the optimum must be determined by experiment for each particular design.

The use of a cylindrical chamber gives a much wider latitude in the squish effect than is possible with a hemispherical chamber, because the depth-to-diameter ratio can be varied to give almost any desired squish ratio. Probably the most successful exponent of the truly cylindrical chamber is the Ricardo Vortex engine as built by several different manufacturers. In this engine the combustion chamber has a diameter approximately half that of the cylinder, while the depth of the chamber — which is governed of course by the compression ratio — is approximately the same as the diameter. The use of the Burt-McCallum single-sleeve valve enables the chamber to be placed in the cylinder head, and thus allows it to be very well cooled and relieves the piston of much of its heat flow problem. An especially valuable feature is the very ample port area given by the sleeve valve. This makes it possible, by suitably shaping the ports, to produce an induction swirl of almost any desired magnitude without the introduction of any interference with the breathing capacity of the engine, and a swirl of such magnitude can be produced that a single-hole nozzle can be used. The swirl is of course augmented by a very powerful squish. The combustion chamber having a diameter only one-half that of the cylinder, three-fourths of the piston area is left for the provision of squish, and in some instances the corner between the open end of the chamber and the flat part of the cylinder head is rounded off for the purpose of moderating the squish. The nozzle is placed so as to deliver the fuel in a direction parallel to the axis of the chamber but at a point near its circumference, as is shown in fig. 102. Instead of being provided with the more usual spring-loaded

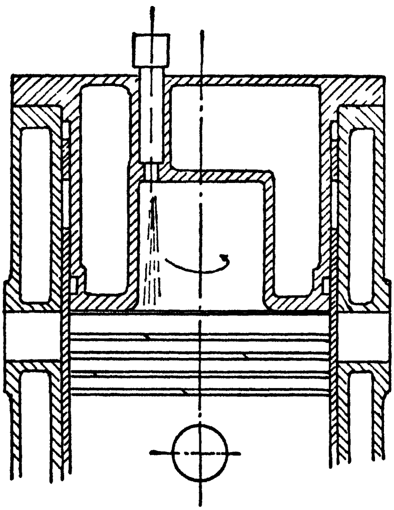


Fig. 102.—Ricardo Vortex chamber

differential valve, the nozzle is of the open type and delivers a comparatively coarse spray, the break-up of which is assisted by the violence of the swirl. The vigorous air movement in this chamber makes it capable of using a very large proportion of the oxygen, a quantity far in excess of that usable with most other forms of open chamber being capable of utilization, a figure of 90 per cent having been reached in at least one instance.

The chamber used by Leyland Motors Ltd. is a development of the cylindrical chamber. Their chamber, shown in fig. 103, is placed in the piston and ordinary poppet valves are used. Instead of being

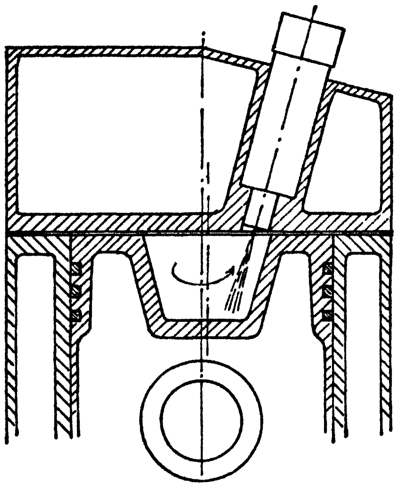


Fig. 103.—Diagrammatic view of Leyland chamber

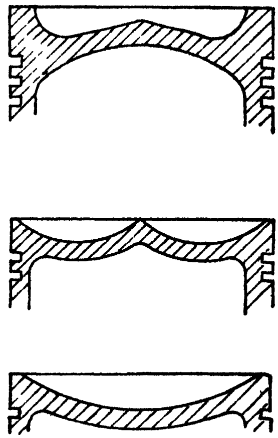


Fig. 104.—Alternative forms of combustion recess for the open chamber

truly cylindrical, however, the chamber is in the form of a truncated cone having its base situated at the crown of the piston, the angle of the cone being about 30°. The corners between the opening and the top of the piston are kept sharp, while there is an appreciable rounding of the corners at the bottom of the chamber. A swirl is produced by the use of a masked valve, the mask extending for nearly half the perimeter of the valve. The nozzle is situated close to the circumference of the chamber, but is placed at an angle with the axis so that the fuel is discharged in a direction substantially parallel to the side of the cone. A single-hole nozzle of a fairly high length-diameter ratio is used, and is closed by a spring-loaded differential valve.

Chambers having a large diameter and shallow depth have frequently been employed, and are the shape most usually adopted for larger engines. The centre of these chambers is frequently provided with a mound, as shown in fig. 104, so as to displace the air from the

centre towards the outside of the chamber and thus make it more readily accessible to the fuel. For high-speed work, however, the lack of squish makes it difficult to obtain really satisfactory results over a wide range of speeds.

4. The Toroidal Chamber.

A form of open chamber which has recently come to the fore is the so-called toroidal chamber, the name being derived from its shape, which approximates to that of a ring or torus. The original idea behind the adoption of such a shape was that, in addition to any other movement which the air might have, a powerful squish would

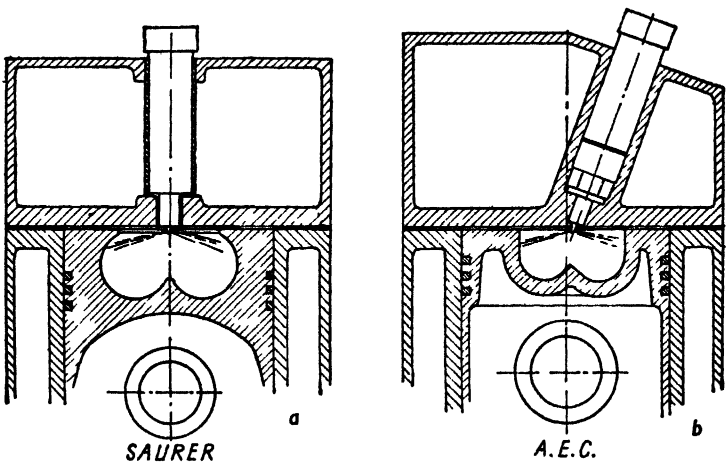


FIG 105.—Diagrammatic representation of toroidal combustion chamber

impart to it a movement similar to that of the familiar smoke-ring. The first exponent of this form of chamber—first at any rate in the field of high-speed engines—was the Tartrais two-stroke engine, and in this engine it is fair to assume that the smoke-ring form of movement was produced because in this engine squish alone was provided. A more recent example of this movement is the Dennis engine, which obtains its air movement from squish alone.

Two other examples are the Saurer (fig. 105*a*) and the A.E.C. (fig. 105*b*), in both of which, however, a rotational movement is provided in addition to the squish; and, as shown in the chapter dealing with air movement, the movement resulting from a powerful squish superimposed upon a vigorous rotational movement partakes of the nature of a vortex, not, as has been supposed, of a toroidal movement accompanied by a rotary motion.

A multi-hole sprayer is used and discharges the fuel rather close

to the mouth of the chamber, a cone angle of some 150° to 160° being employed. In the case of the Saurer and A.E.C. engines four holes are used, while the latest form of the Dennis engine has six. In a smaller engine of this type the Saurer Company have used an outwardly opening nozzle designed to give a spray in the form of a thin sheet across the whole mouth of the combustion chamber, the air flowing through this sheet of spray as it enters the chamber. It is understood that no swirl is used with this form of sprayer, squish being employed alone, so that the air movement here is of a toroidal nature. The great advantage of this form of chamber is the very powerful squish that can be obtained. The use of a powerful squish necessitates only a moderate amount of induction-induced swirl, and only a small mask is therefore needed on the inlet valve, reducing to a minimum the resistance offered during the influx of air. At the same time, the swirl is built up to the intensity required for optimum results at the moment it is required, that is, during the injection and combustion periods. There is thus no time for it to die down between the point at which it is generated and that at which it is required, and the engine is thereby enabled to maintain its best performance over a wider speed range than when the swirl is built up wholly during induction. Further, the ability to build up a more vigorous swirl than could otherwise be obtained enables a higher proportion of the oxygen to be utilized effectively without too high a price being paid in increased mechanical losses incurred in producing the swirl.

A drawback common to practically all open types of chamber is the need to employ a high injection pressure. In order to obtain a fine state of atomization of the fuel during the early stages of injection, and so reduce the delay period, a high initial velocity of injection is necessary. This demands a high opening pressure for the nozzle valve, which influences the minimum quantity of fuel that can be delivered per injection and introduces complications at light loads and under idling conditions, a feature which has particular reference to vehicle engines. Further, the small area of the orifice necessary with most engines involves very high injection pressures at high engine speeds and results in a large amount of distortion taking place in the injection as the engine speed changes. This is a matter of little importance for constant-speed engines, because the injection equipment can be selected to suit the designed speed; but in the case of variable-speed and variable-load engines it becomes a matter of first importance and has a direct bearing upon the range of speeds over which the optimum performance can be maintained. The ultimate result is that the air movement and the rate of fuel delivery get out of step and the efficiency of the engine suffers in consequence.

It might be argued that some of the forms of chamber referred to above can hardly be considered as open types of chamber, in that the

air is transferred from the cylinder into what is really a separate chamber. There is perhaps some justification for such an argument, especially in the case of some of the toroidal chambers. It is of course difficult to draw an exact line of demarcation between an open combustion chamber and some of the forms of separate combustion chamber. The dividing line is really a question of the size of the opening between the cylinder and the combustion chamber proper. When the size of the connexion between the two is small compared with the diameter of the cylinder, then the chamber may be considered to be some form of separate combustion chamber; but when the size is fairly large with respect to the cylinder diameter, then the chamber may be considered as being an open type of chamber. Generally speaking, the mouth of the open chamber will not have a diameter much less than half that of the cylinder, whereas any form of separate combustion chamber will, as a rule, have a passage between itself and the cylinder with a diameter certainly not greater than one-fourth of that of the cylinder, and usually much smaller than this.

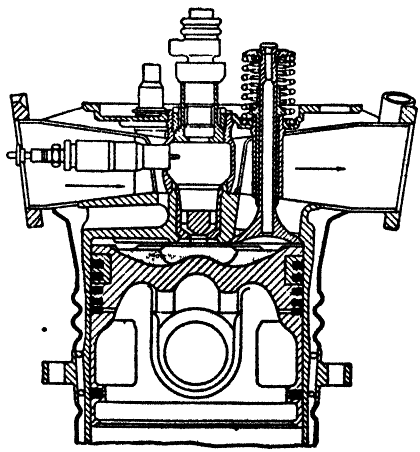
5. The Pre-combustion Chamber.

The underlying motive of the pre-combustion chamber is an attempt to utilize the combustion of a small portion of the fuel to produce an action similar to that of the air blast of the Diesel engine. To this end the combustion chamber is divided into two parts—the main chamber, which forms an extension to the cylinder proper and contains the major portion of the air, and the pre-combustion chamber, which is separated from the main chamber by an orifice, or orifices, of relatively small area. The main chamber is usually formed by a shallow depression in the piston, while the pre-combustion chamber takes numerous different forms and is situated in the cylinder head, and is either wholly or partly uncooled. The whole of the fuel is delivered directly into the pre-combustion chamber and in such a manner that the bulk of it will reach the neighbourhood of the orifice separating the two chambers, so that when ignition takes place in the pre-combustion chamber the rise in pressure ejects the fuel into the main chamber wherein the bulk of the combustion takes place. The orifice between the two chambers is designed to act as a distributer and spread the fuel throughout the main chamber, the burning and partially burned fuel particles being injected in a number of streams in such a way as is thought to search out the main chamber in the most effective manner. This form of chamber is very popular in Germany but has never attained any great popularity in this country, although a fair number of imported engines of this type are in use here.

One of the earliest engines of this type was the Bronz engine built in Switzerland. In this engine the fuel ran by gravity through a suitable valve into the pre-combustion chamber during the latter

part of the suction stroke, and ignition followed as soon as the compression temperature, assisted by the heat retained by the chamber, reached the necessary figure. As will be imagined, the lack of control over the precise moment of ignition was a serious drawback, and these engines, which were exceedingly heavy in order to take care of the high maximum pressures, were limited to speeds in the neighbourhood of 800 r.p.m. and barely came within the category of high-speed engines.

This chamber has been developed with considerable success by the Daimler-Benz A.G., who have employed it for a great number of years and have recently applied it to engines of large power running at a speed of 2400 r.p.m. and intended for airship propulsion. The engines produced by this firm have a normal injection system utilizing a single-hole pintle type of nozzle. The pre-combustion chamber is elongated and roughly pear-shaped in form, with the orifice at the narrow end and the injector nozzle at the opposite end. The form of the chamber, which is designed to contain about one-third of the compression volume, has undergone a number of changes during the years of development, as has also that of the orifice or "burner". The latter is modified according to the position occupied by the pre-combustion chamber, this having been placed vertically over the centre of the cylinder in many of the earlier engines; but in some of the later models, in the interests of the easy accommodation of valves of a size suitable for high-speed operation, it has been given an inclined position towards one side of the centre of the cylinder. The form of this chamber, as applied to the airship engines, is shown in fig. 106.



By courtesy of The Automobile Engineer.

Fig. 106.—Daimler-Benz circulation chamber as applied to their airship engine

6. Swirl-chamber Engines.

The term "swirl-chamber engine" is synonymous with "compression-induced swirl engine", and being shorter and perhaps somewhat more descriptive of the form assumed by such chambers, is to be preferred. Basically this type includes a combustion chamber separated from the engine cylinder, and into this chamber as much of the air as possible is transferred during the compression stroke. The form of

the chamber is one calculated to assist in the production of a rotational movement of the air within it, and it is connected to the cylinder by means of a throat or passage which enters the chamber in a tangential direction so that the air on flowing into the chamber is given a rotary movement within it. The fuel is injected into the swirl chamber, and the ignition and the bulk of the combustion take place therein. The rise of pressure following upon ignition forces the gases out into the cylinder again as the piston moves outwards on the working stroke. By the use of this chamber a very powerful swirl can be produced, and this enables a single-hole nozzle to be used, the type commonly adopted being some form of pintle nozzle. The compact nature of the chamber and the high rate of swirl have a number of advantages: a high percentage of the available oxygen can be used without running the risk of having a dirty exhaust, the vigorous nature of the swirl makes combustion conditions to a great extent independent of the state of the nozzle, and troubles from exhaust smoke due to faulty nozzles are rare. Being dependent upon engine speed, the air swirl maintains a rate proportional to the engine speed at all rates of revolution. The high rate of swirl makes it possible to use a relatively low injection pressure in conjunction with a nozzle orifice of appreciable size, a nozzle-opening pressure of around 100 atmospheres being all that is necessary. This figure—which is about half that required for an open combustion chamber—and the large orifice area of the pintle nozzle result in only a small amount of distortion of the injection period as the speed changes and make it possible to retain a greater measure of control over combustion conditions than is obtainable with some other forms of combustion chamber.

The transference of the air to a separate chamber makes it possible to provide a heated surface over which the air is caused to flow during its passage from the cylinder to the combustion chamber. By this means the temperature of the air at the end of compression is raised above that which would be reached by the same degree of compression unaided. This increase in compression temperature makes the engine much less sensitive to changes in fuel characteristics, with the result that it can use satisfactorily fuels of a lower ignition quality than engines having some other forms of combustion chamber. At the same time, the higher temperature of the chamber walls reduces the quantity of products of partial combustion and gives a less odorous exhaust.

These advantages are obtained, however, at some sacrifice in fuel economy. The work done during compression is fairly considerable, and there is a corresponding loss during expansion also, with the result that the mechanical efficiency of the engine suffers; and although the thermal efficiency based upon the indicated horse-power is of a very high order, that based upon the brake horse-power is rather less than can be obtained with the open combustion chamber. The high

degree of air utilization, however, enables a high brake mean effective pressure to be developed.

In some designs an attempt is made to reduce the transfer loss by increasing the size of the passage between the cylinder and the combustion chamber, and introducing an obstruction into the opening to reduce its area towards the end of the compression stroke in order to increase the gas velocity, and so to obtain the desired swirl velocity; but, as explained earlier, the total amount of work remains the same and no practical advantage is gained by the device. In some designs definite disadvantages result from this device and in no instance does the net performance of the engine appear to benefit.

The swirl-chamber engine is not actually a development of the high-speed engine. It dates back to a time when even petrol engines did not reach speeds much in excess of those now considered as "medium" for compression-ignition engines. Its present form, however, is mainly due to the work of Ricardo, who has demonstrated how to take advantage of the opportunities it offers.

7. Types of Swirl Chamber.

Two forms of swirl chamber are illustrated in fig. 107; at A is shown a chamber which has been employed quite extensively in research work and which gives a toroidal movement to the air. In the author's

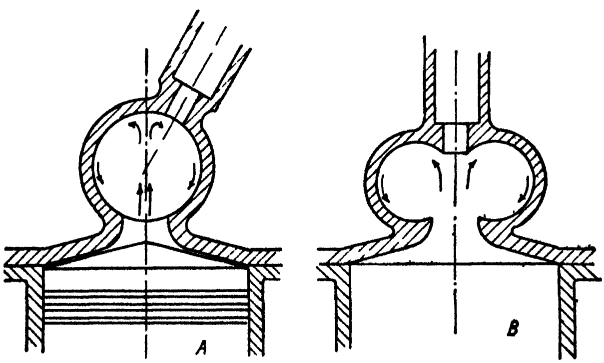


Fig. 107.—Forms of swirl chamber

opinion this is not well adapted to produce an ordered movement of the air, because after passing across the diameter of the chamber the air strikes squarely against the opposite wall instead of being guided smoothly around the chamber walls.

Another form which does not suffer from this defect is shown at B, but to offset its advantage in this direction the spray path is apt to be somewhat limited. This latter chamber is used with an obstructed

orifice, but it will be clear that it could also be made without any obstruction, while an obstruction could be as readily employed with the form shown at A.

Various forms of simple swirl chamber, in which the air revolves about a single axis, are shown in fig. 108.

That shown at A in fig. 108 is similar in principle to that shown at B, but has the advantage that the chamber does not project so far beyond the cylinder bore, although the amount of this projection can be reduced by placing the throat vertically as at C, or by moving the chamber so that the throat comes nearer to the centre of the cylinder

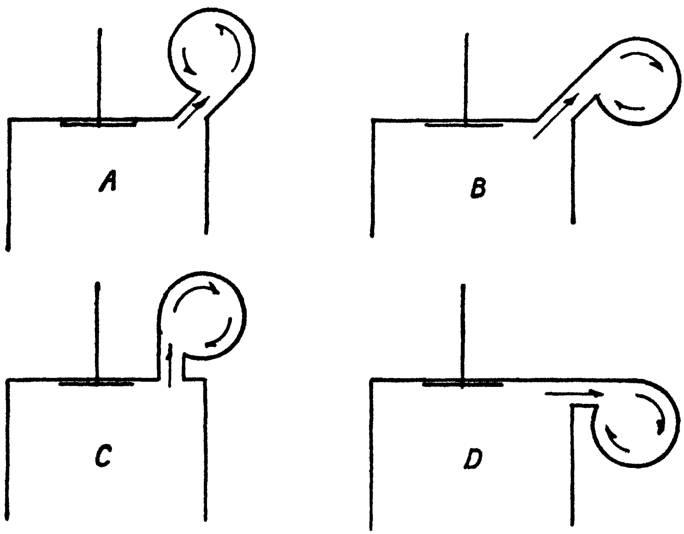


FIG. 108.—Showing chamber giving a simple swirling motion

as in the Perkins engine. A vertical throat, however, is not quite so good from the point of view of the flow of gas to and from the cylinder, the inclination of the throat serving to guide the gases and search out the air which remains in the clearance between the piston and the cylinder head in a way not possible with a vertical throat.

All these three chambers have unobstructed passages, and, in fact, the inclined passage does not readily lend itself to the effective introduction of an obstruction. With a vertical passage an obstruction can be more easily provided.

Fig. 108, D, shows a form of chamber adapted to an obstructed passage. In this instance the piston itself closes up the opening towards the end of the compression stroke by overrunning the opening; no projection is necessary on the piston, and this can therefore be made symmetrical. In this design the swirl chamber can be accommodated

either wholly in the top of the cylinder casting or with the passage and lower part of the chamber in the cylinder casting while the rest of the chamber is placed in the cylinder head. The detail variations of these several types are, of course, numerous and many of them have been the subject of patents.

The swirl chamber just illustrated may be either spherical or cylindrical according to the designer's ideas; if the cylindrical form is chosen, it is usual to make the length of the cylinder less than the diameter, somewhere in the region of one-half, and the fuel is almost always injected either radially inwards or with a slight " downstream " bias.

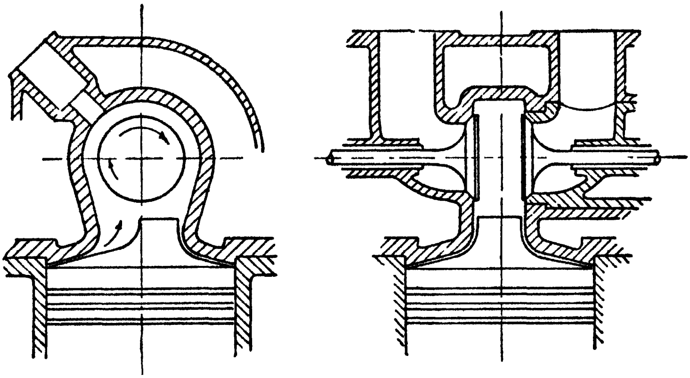


Fig. 109.—Showing the " clerestory " combustion chamber

A variant of the spherical chamber is a flattened sphere, i.e. a chamber which in one view is truly circular, while a section at right angles to the first is oval in form.

If desired, any of these chambers may be provided with more than one opening connecting them to the cylinder, and the openings may take one of many forms, such as circular, oval, rectangular, rectangular with rounded ends, or kidney-shaped in cross-section, and may be either parallel or tapered in their length.

In all of the above types the valves are situated in the unoccupied portion of the cylinder head and perform their function entirely independently of the combustion chamber.

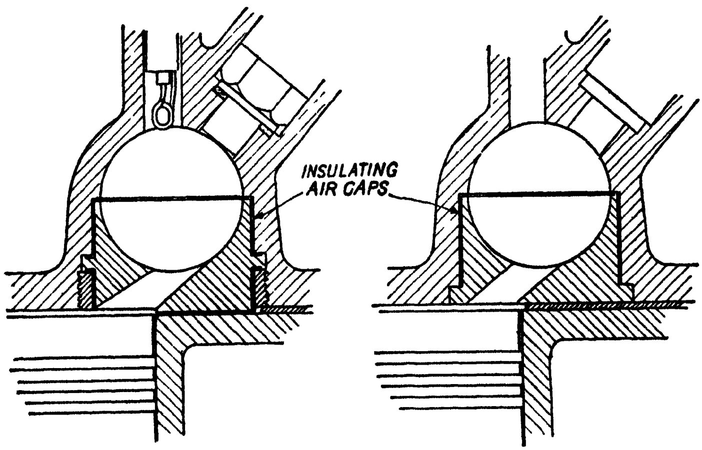
A form of chamber which has frequently been adopted is shown in fig. 109. This has the valves situated in the chamber itself and of necessity, therefore, has an opening between the cylinder and the chamber which is large enough to pass the charge during the induction stroke without excessive throttling. Such chambers are usually cylindrical in form with the valve heads arranged so as to form the ends of the cylinder, although, by suitably shaping the heads of the

valves, the chamber could be given a spherical form or even a "corpuscular" shape if considered advantageous. To obtain the requisite degree of swirl the piston is provided with a knob which partially closes the opening at the end of the compression stroke; these knobs assume a variety of shapes in order to allow the incoming air to be directed in varying ways during the final stages of compression. The injector must be mounted radially, but there is considerable latitude as to its position relative to the opening, and the spray, or sprays, may be varied as to direction in quite a number of ways.

It will be seen from the above that the swirl hamber for an engine using a compression-induced swirl may have an almost infinite variety of form. A great advantage enjoyed by many forms is the possibility of introducing an uncooled surface to assist in raising the compression temperature during the actual compression stroke, and so avoiding any loss of volumetric efficiency by heating the air during induction. The provision of an uncooled surface in an oil-engine combustion chamber for the purpose of assisting in obtaining ignition is almost as old as the oil engine itself, certainly as old as the principle of fuel injection, because it was used for the original Akroyd-Stuart engine, but the present-day application of the scheme must be credited to Ricardo. In its original form the Ricardo swirl chamber was spherical, and was situated in the cylinder head towards one side of the cylinder. Only the upper half of the chamber formed part of the actual cylinder head casting, the lower half being formed by a separate part made of a heat-resisting material, and fitted into place in such a way that the flow of heat from this part to the cylinder head casting, and so to the cooling water, was made as difficult as possible. The passage from the swirl chamber to the cylinder is cut in the uncooled portion, so that the air, when flowing into the chamber during compression, picks up heat to assist in cutting down the delay period and speeding up the process of combustion. During the expansion stroke the reverse action takes place. The burning gases flowing out through the passage restore the heat extracted during the inward journey and so maintain the hot body at a high temperature. The heat transfer does not depend solely upon the passage itself, because the air rotating within the chamber is in contact with the hot surface of the lower half of the sphere, as will be seen from fig. 110.

The air velocity through the passage is of a fairly high order, although not nearly so high as might at first appear, because during the later stages of compression, when the rate of transfer of air to the swirl chamber is a maximum, the density of the air is high and its volume much reduced. Conditions are excellent for a high rate of heat transfer between the air and the metal, and the exchange of heat increases as the engine speed increases because, despite the fact that the time factor decreases as the speed increases, the temperature of

the hot surface increases with the quantity of fuel burned in the chamber, i.e. with the speed, and at the same time the increased rate of flow over the surface increases the rate of heat exchange. The influence of the heat picked up in this way is a very real one, as is shown by the figures given in Tables XVI and XVIII (pp. 131, 137), which show the increase in compression temperature as both load and speed are increased. The increase in compression temperature with speed is a very valuable feature, because it assists in reducing the delay period and thus helps to maintain the combustion conditions constant throughout the speed range. Contact between the fuel and



Ricardo method of construction A.E.C. method of construction

Fig. 110.—Showing the Ricardo "Comet" swirl chamber and alternative methods of securing the hot plug

the hot surface, the temperature of which increases with speed, may also play a part in the reduction of the delay.

To minimize the flow of heat directly from the hot body to the water-cooled material of the cylinder head, the actual contact between the two is made as small as possible, and is confined to a narrow flange at the base of the hot body; at all other points a small air gap a few thousandths of an inch wide is left between the two and so insulates the hot body. Under operating conditions the gap gets filled with carbon, which, being a poor conductor of heat, does not materially alter the heat conditions but has the advantage of transferring the air in the gap to a more useful position in the chamber. In insulating the hot plug from the cylinder head, Ricardo himself takes extreme measures; he not only reduces the size of the connecting flange to a minimum but undercuts the flange itself to reduce the area of contact yet further. A ring nut is used to screw the plug in place, as is shown

in fig. 110. An alternative method, used by the author's firm, also shown in fig. 110, has the advantage of being somewhat simpler from a manufacturing point of view. This is to make the narrow flange a close fit with the cylinder head but make all other parts with a small air gap; the underside of the cylinder head and the surface of the hot body are then ground flush with each other, and the two parts then become virtually a single part with a flush surface against which the cylinder head gasket will fit. The hot plug, being made of heat-resisting steel, has a very poor heat conductivity, and the flow of heat from it to the cylinder head is thereby restricted.

The question of the conductivity of the material from which the hot plug is made is one of some importance. It has been argued that

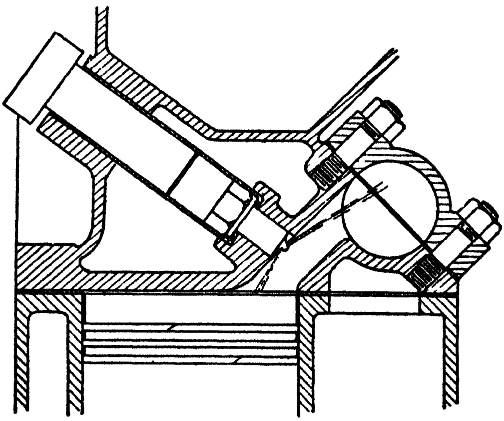


Fig. 111 —Perkins swirl chamber

with a poor conductivity the plug would attain a really high temperature, and therefore produce a further reduction in delay. Experiments were therefore tried with plugs made in refractory materials, but with disappointing results; the delay was apparently increased, as the engine was noticeably less smooth than one fitted with the usual heat-resisting materials.

On the other hand, the author has observed that hot plugs which during experimental work had been made from ordinary mild steel (the conductivity of mild steel is several times that of a heat-resisting steel) invariably resulted in slightly smoother running than those made in heat-resisting materials. As an experiment, therefore, some plugs made of aluminium bronze were tried and showed a further slight improvement in smoothness. From these experiments it may be concluded that, with a plug made of a material having a poor conductivity, the flow of heat from the interior of the metal to the air is restricted and the quantity of heat picked up by the air is therefore reduced.

With a high conductivity the flow of heat takes place more readily, and the compression temperature is increased accordingly. Unfortunately, high-conductivity materials have only a short life, too short to allow them to be used commercially.

Chambers are also used in which it is the upper half that is uncooled. An example of this type is the Perkins chamber, which is illustrated in fig. 111. The chamber in this design is a short cylinder or "cheese" shape, and the fuel is introduced by a sprayer fitted in the connecting passage and delivering the fuel through a two-hole nozzle both up the passage into the swirl chamber and down the passage towards the working cylinder.

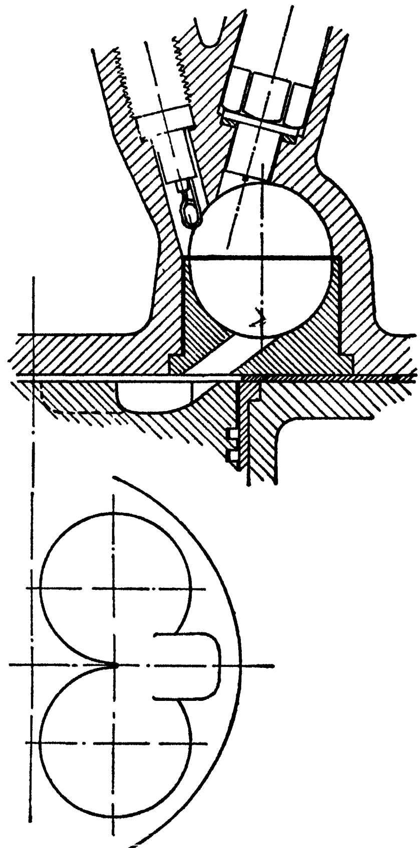


Fig. 112.—Ricardo Comet Mark III combustion chamber showing plan view of recess in piston

A later form of the Ricardo chamber known as the Mark III uses a combination of a swirl induced during compression and a swirl induced during combustion. In this type only about 50 per cent of the air is transferred to the swirl chamber, which, however, is precisely similar in form to that of the earlier chamber. The remainder of the air is transferred at the end of compression into two shallow circular depressions in the piston crown; these are arranged so as to intersect, or nearly so to do, and are provided with a third short depression which serves to connect the intersection of the two circular depressions with the opening to the swirl chamber and forms a tangent to each of the two circular depressions (fig. 112). The whole of the fuel is delivered into the swirl chamber, and on ignition taking place the increase in pressure in the swirl chamber causes a rush of burning gases and partially burned fuel outwards into the two shallow depressions in the piston via the connecting passage. A rotating movement is thereby set up in the depressions, and the air contained in them is

fed into the gases which are streaming out of the swirl chamber and so helps to complete the combustion.

The advantage of this system over the original form is that much less work is done in transferring the air from the cylinder to the swirl chamber and back again; only half the air is transferred, so that not more than half the work is necessary. In addition, the air between the piston and the cylinder head, which in the older type of chamber is in a thin layer and not therefore readily available to fuel issuing from the swirl chamber, is now partly embraced by the shallow depressions which cover a quite appreciable part of the surface of the piston. The proportion of the air contained in the clearance space between the cylinder head and the piston is thus reduced, and more of it is made readily available for receiving fuel. This arrangement gives a maximum smoke-free B.M.E.P. some 10 per cent greater than can be attained with the older design, together with a corresponding improvement in fuel consumption on the brake horse-power at all loads and speeds, although the consumption on the indicated basis is actually somewhat lower on account of the rather later burning of the air outside the swirl chamber.

Engines which have the valves opening directly into the swirl chamber are sometimes provided with a lining around the throat of the passage arranged so as to have an effect similar to that of Ricardo's hot plug. The effect is assisted by the knob on the piston provided for the purpose of reducing the area of the passage. This knob reaches a fairly high temperature, and the rapid flow of air over it during the final stages of compression results in some exchange of heat. The presence in the chamber of the hot exhaust valve head serves the same end, although the intimacy of contact being less, the rate of exchange of heat will not be so great.

One drawback of the swirl chamber using a high rate of swirl is that, under starting conditions when the engine is cold, a high loss of heat is sustained, making starting somewhat less easy than in the case of the open combustion chamber. To overcome this difficulty an extraneous source of heat is employed in the shape of a " glow " plug. This plug is heated electrically and serves not as a source of heat to raise the temperature of compression—the amount of energy supplied is much too small for that—but as an incandescent point which will ignite any fuel striking against it. The only real objection to the use of such a device is that it is an extra piece of equipment to be maintained, but its presence insures a start under temperature conditions which would make a start impossible with the open chamber. Generally speaking, such a device or its equivalent is necessary in nearly every type of chamber other than the open type.

8. The Air-cell Chamber.

The air-cell combustion chamber is normally associated with a combustion-induced swirl, although in most cases the air movement is more of an indiscriminate nature than that of an orderly swirl. Here, also, the combustion chamber is divided into two parts, one directly connected with the cylinder bore and the other separated from the first by a restricted orifice. This orifice is situated in an insert, made of a heat-resisting material on account of the high temperatures to which it is subjected. The fuel is not introduced directly into the secondary chamber, but is injected into the main chamber in such a way that it passes across the main chamber and is aimed to strike at the orifice connecting the two chambers.

Much discussion has taken place as to the precise action of these chambers, the point at which ignition originates, and whether or no combustion actually takes place inside the air-cell. From personal experience with this type the author is of the opinion that the behaviour differs with different types of engines. In the Acro chamber it would seem that little or no combustion takes place actually inside the cell, but that a good deal takes place around the orifice between the two parts. In the form of this chamber with which the author is most familiar the air-cell was uncooled, and under normal conditions of operation showed no visible signs of a red heat in daylight and only a very dull red in a darkened room, thereby indicating that very little, if any, combustion could be taking place inside. A sticking nozzle, however, or an excessively early timing of the fuel injection, either of which would cause a quantity of fuel to enter the air-cell, invariably resulted in the cell becoming incandescent and, on occasion, being completely destroyed.

From such evidence it appears conclusive that in this particular engine combustion does not normally take place inside the air-cell. On the other hand, in the case of the most recent example of the air-cell engine, the Lanova, there is definite evidence that quite a respectable measure of combustion takes place inside the cell, and the patentees of the scheme claim specifically that combustion originates within the cell and results in the ejection of its contents into the main chamber, where a definite swirling movement is produced by suitably shaping the chamber.

In the original air-cell designs it was claimed that the air in the cell was brought into play after combustion had started, and the fresh supply of air thus derived enabled the combustion to be completed without smoke. To operate in this fashion, the air from the cell would not be received, and burned, until late in the cycle, and the efficiency attainable from its use would be of a low order; and in actual fact the efficiency of these earlier types, although perhaps good by comparison

with the petrol engines with which they had to compete, was decidedly poor by modern standards.

The Lanova chamber (fig. 113) reverses the procedure of the earlier air-cell engines. The fuel is deliberately discharged into the air-cell, although indirectly and via the main chamber, and ignition takes place inside the air-cell, ejecting the contents thereof into the main chamber. In plan the main part of the chamber is shaped something like the figure eight, the two circular parts of the figure eight lying beneath

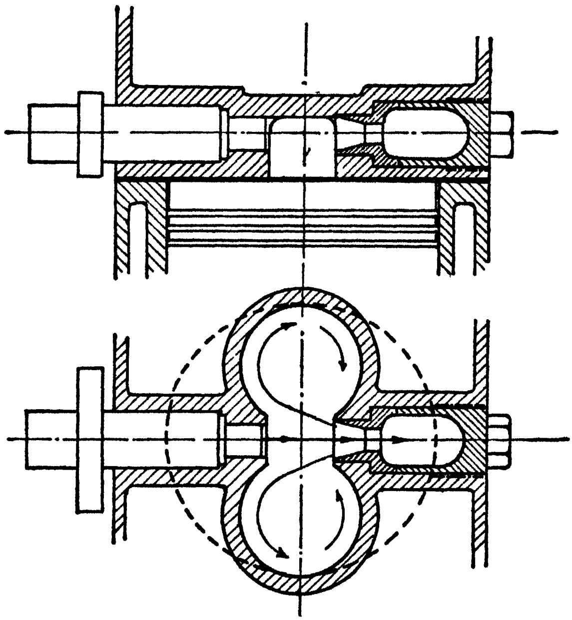


Fig. 113.—Diagram of air-cell chamber of the Lanova type

the valves, which are placed in the cylinder head. The air-cell is placed horizontally and connects with the main chamber at the " waist " of the eight, while the fuel nozzle is situated on the far side of the eight immediately opposite to the air-cell. During injection the fuel is discharged across the main chamber and into the mouth of the air-cell. The air-cell is only partially cooled, and the higher temperature in the cell results in ignition taking place therein, and causes the discharge of the contents back into the main chamber against the stream of fuel particles still issuing from the nozzle. The rush of gases from the air-cell imparts a rotary motion to the air in the outer chamber, the air in the two loops of the figure eight rotating in opposite directions and bringing the air into contact with the fuel stream.

With this chamber it is claimed that only a comparatively low compression pressure, about 350 lb./sq. in., is necessary, that the maximum pressures are correspondingly reduced, and that brake mean pressures of a high order with a clear exhaust are obtained. To facilitate starting with the low compression ratio, the air-cell is divided into two parts, one of which may be closed off by means of a valve provided for the purpose, thereby raising the compression ratio for easy starting.

In the air-cell engine only about one-third of the air is transferred to the cell, so that the work done during compression is not very great and the mechanical efficiency should therefore be of a fairly high order.

The fact that the combustion-induced swirl bears little or no relationship to the speed of the engine makes it difficult to obtain the optimum results over a wide range, and in this country, when a fairly wide speed range is demanded—as is the case for vehicle work—the air-cell engine does not now command much interest.

The enormous amount of development work done for, and the very large amount of experience gained from, vehicle engines during the last decade is now being drawn upon for the design of combustion chambers for the faster running engines of all types, and as a result, higher mean pressures are being obtained than before were considered possible.

In the vehicle engine field, the higher economy obtainable with the open chamber, coupled with the heavy taxation on fuels used for road transportation, has recently diverted attention from the other types and caused a concentration upon the development of the open chamber, and has resulted in an improvement in its performance over a wider speed range.

On the other hand, the advantages of the swirl-chamber engine have been seized upon by some of the builders of slower speed engines, who realize that ability to use a wide range of fuels is a very real advantage, especially as in many localities fuels of relatively poor ignition quality command a lower price, so that any sacrifice in fuel consumption is compensated for by the lower cost of the fuel. The swirl chamber is now being used for engines designed to run at speeds which hardly justify the term "high-speed"; thus, although the high-speed engine can in no way be considered to be a development from the low-speed engine, the development of the latter is being assisted by the high-speed engine.

In this country, although during the early stages of development of the high-speed engine the air-cell chamber and the pre-combustion chamber commanded a certain amount of attention, this soon disappeared, and interest centred around the swirl chamber and the open combustion chamber types, both of which, incidentally, are essentially British developments. The outstanding advantages of

these two types, the high economy of the open chamber and the flexibility and more catholic taste in fuels of the swirl chamber, quickly outweighed the fact that both the other types had already undergone a considerable amount of development elsewhere before being introduced here, and, lately, the swirl chamber, and to a lesser extent the open chamber, have been gaining ground on the Continent in the strongholds of the air-cell and pre-combustion chamber.

CHAPTER IX

Fuel Injection

1. The Requirements of Engines for different Services.

While there are features of the fuel injection system which are common to engines of all types, there are essential differences in the requirements for different engines, which arise from the nature of the load which the engine is designed to carry, or, more accurately, from the way in which the load varies under service conditions. Engines can be classified under three headings, according to the nature of their loads:

(1) Stationary Engines.
(2) Marine or Aircraft Engines.
(3) Vehicle Engines.

The stationary engine is mainly a constant-speed machine, the total speed variation being not greater than the limits within which the governor, with which such engines are almost always provided, can hold the speed between no-load and the maximum overload carried. This variation does not normally exceed 5 or 6 per cent of the nominal speed, and at the most 10 per cent will cover it. It is true that engines of any given design may be called upon to operate at different speeds according to the requirements of a particular installation, but under such conditions any adjustment necessary to the injection equipment in order to meet such changes in speed will be made at the factory and will be of a permanent nature. In a few instances engines may be called upon to " tick-over " at a speed much below their normal running speed before being placed in service, but such conditions do not usually involve any question of economy and do not therefore introduce any complications beyond the requirement that such periods of " tick-over " must not result in fouling the spray nozzle.

Marine and aircraft engines are coupled to a screw propeller which imposes upon the engine a definite load-speed curve from which there is no departure unless the dimensions of the propeller are changed. The aircraft engine, it is true, is subject to rather more latitude than the marine engine, because of the variation in the density of the air

with altitude. This variation influences the power of the engine as well as the resistance offered by the propeller, and is a condition peculiar to aeronautical work, but does not really influence the fuel injection system beyond the necessity for restricting the fuel supply according to the quantity of air received by the engine, while the variable-pitch propeller provides a convenient means for maintaining a constant engine speed.

The power absorbed by a screw propeller, whether running in air or water, varies as the cube of the rotational speed, so that once the propeller has been selected the load on the engine and the speed are connected by the relation B.H.P. $\propto$ (r.p.m.)^3, from which it follows that B.M.E.P. (or torque) $\propto$ (r.p.m.)^2.

The vehicle engine is called upon to operate under conditions which differ greatly from those for the other two classes; speed and

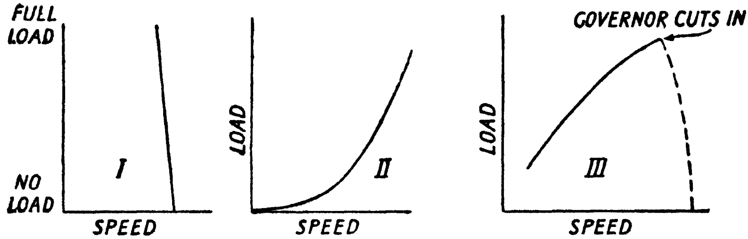


Fig. 114.—Load/speed diagrams

load bear no relation to one another but vary quite independently. The limits are imposed, on the one hand, by the maximum output that can be obtained from the engine at any given speed within its operating range and, on the other, by zero output at all speeds within the same range. Compression-ignition engines for vehicle work are now almost universally provided with a maximum speed governor which limits both the maximum output and the maximum speed. The lower end of the speed range is usually limited by the operator, owing to the necessity for keeping a reasonable engine speed for ease of handling and to avoid stalling the engine.

The conditions to be fulfilled by the three classes can be best illustrated graphically by a load/speed diagram as shown in fig. 114. In this figure, I represents the conditions under which a stationary engine works: the load varies along a line, very nearly straight, connecting the full load speed and power with the speed at which the engine runs under zero load. The marine engine is shown in II, the load and speed being connected by a curve, the equation for which is B.H.P. $= f$(r.p.m.)^3. The vehicle engine (III), on the other hand, has to cater for an area, the area being the space enclosed by the speed/power curve

at maximum output within the engine's speed range, the line representing zero power at all speeds up to the maximum, and the line along which the power falls away to zero when the governor comes into operation. The load on the engine will vary from moment to moment from any one point to any other within the space enclosed by these three lines, and does so frequently and continuously.

The fluctuations of load and speed are much wider and more frequent in the case of a vehicle engine than for either of the other two classes. In fact, they are continuously in a state of variation as road and traffic conditions change. Stationary engines are on occasion subject to fairly rapid fluctuations in load, but only between the narrow limits of speed set by the governor; thus the conditions to be fulfilled are much simpler than for the vehicle engine. The marine engine has perhaps the simplest running conditions because, except when manœuvring in the neighbourhood of a port, it is usually called upon to operate at a steady load and speed for prolonged periods.

It will therefore be seen that the fuel injection equipment has widely differing conditions to meet according to the service demanded from the engine. The injection equipment must be capable of delivering the fuel to the engine in such a manner that the best results can be obtained at all points within the operating range. Clearly this is easiest to fulfil for Class I; the speed is constant and the injection equipment has only to vary the quantity of fuel to meet the demand. It is true that the condition in which the fuel is delivered to the engine must be equally suitable whether the charge is large or small, but this is a relatively simple matter in the absence of disturbing influences from wide changes in speed. For Class II a rapidly decreasing charge is required as the speed decreases, but for any given fuel charge the speed is always the same.* The injection equipment for Class III, however, must be capable of delivering any quantity of fuel up to the maximum at any speed within the working range of the engine, in a manner such that the engine is capable of giving its best performance under every circumstance. This must be done without any adjustment made by the operator because (1) the variations in conditions take place so rapidly that it would be quite impossible for even the most skilled operator to adjust matters so that the best possible results would be obtained, and (2) engines of this class are usually placed in the hands of unskilled labour. The conditions called for by Class III are thus far more arduous and difficult of fulfilment than for either of the other two classes, and it will easily be understood that a system admirably suited to Class I or II may be totally unsuited to Class III.

* Some deviation from this will be found when a vessel is used for towing.

2. The Requirements from the Injection System.

As we have said, the requirements from the fuel injection system are, in brief, that it shall deliver the fuel in the manner best suited to the engine's needs, and do this at any load or speed at which the engine may be called upon to operate. This means that the fuel must be delivered at such a rate, at such a period in the cycle, and in such a condition that the requisite output is attained at the maximum possible efficiency, after taking into consideration such limitations as it is necessary to impose in order to restrict the maximum pressure and ensure that the requisite degree of smoothness of operation is obtained. Bound up with the question of efficiency is the question of cleanliness of the exhaust, and to some extent its freedom from obnoxious odours. Lack of cleanliness in the exhaust means unburned, or partly burned, fuel and therefore a loss of efficiency, while an evil-smelling exhaust is mainly caused by imperfect combustion and therefore implies some loss in efficiency, although in this instance the loss may be exceedingly small. Sulphur in the fuel may give rise to smell without any imperfection in the combustion.

It has already been shown elsewhere that in order to obtain a high efficiency certain conditions as to the addition of heat, i.e. rate of combustion, must be fulfilled. These conditions must be maintained at all speeds in the case of an engine called upon to operate at widely differing speeds, which means that, somehow or other, the rate of burning of the fuel must be made to vary with the speed of the engine. The achievement of this result is largely dependent upon conditions within the engine itself, but before a high efficiency can be attained by the engine, the injection equipment must first deliver the fuel in the right quantity at the right time and in the right state.

3. Types of Injection Equipment.

Injection equipments can be divided into two general classes, air-blast injection and mechanical injection, sometimes called "solid" injection. The former, however, is not used for high-speed engines and has practically disappeared from the low-speed field, and it therefore will not be further discussed here except to state that it was the advent of satisfactory mechanical injection equipment that made the high-speed engine possible.

In all forms of mechanical injection the fuel is subjected to high pressure and is forced into the combustion chamber through a special device for converting the fuel into a fine spray in order to make it capable of rapid and complete combustion.

The various forms of mechanical injection equipment may be divided into three classes:

(1) The Constant Pressure or "Pressure Rail" System.
(2) The Spring Injection System.
(3) The Cam-operated Plunger or "Jerk Pump" System.

All three systems have been used with satisfactory results, but they are not all equally suitable for all three classes of engine. Their characteristics differ materially.

4. The Constant Pressure or "Pressure Rail" System.

In this system the fuel is maintained at a predetermined pressure by means of some form of pump and some form of hydraulic accumulator. The speed and phasing of the pump need have no relationship with the crankshaft and may, if need be, be driven independently of the engine it is designed to feed. The only condition imposed is that the required intensity of pressure is maintained at all times. The fuel is delivered to the engine by means of a mechanically-operated fuel valve driven from the engine camshaft and suitably timed thereby. The quantity of fuel delivered to the engine is governed by the lift, area, and duration of opening of the valve and the pressure of the fuel in the system. The fuel valve may be either of the spring-loaded poppet type operated by suitable means from the camshaft, or of the "flash" variety which places the spray valve in the cylinder head into communication with the pressure in the system for the desired period of time and then cuts it off again.

5. The Spring Injection System.

In this system the pressure on the fuel is built up by the compression of some form of spring, the load from the spring being applied in one of two ways. It may be applied directly to a form of hydraulic accumulator and the fuel delivery be controlled by a flash valve, or the load from a precompressed spring may be applied suddenly at the required moment to a plunger by means of a suitable trip gear, the termination of injection being controlled by a spill valve or other suitable device. In the former case the system really becomes a variation of system (1).

6. The Cam-operated Plunger or "Jerk Pump" System.

In this system the pressure on the fuel is built up by means of a plunger operated by means of a cam or eccentric. The movement of the plunger is suitably phased with the crankshaft to ensure that fuel delivery takes place at the correct period of the cycle, and the actual timing and duration of the injection are controlled by a suitable flash valve. The quantity of fuel delivered is regulated by varying either the stroke of the plunger itself or the effective stroke of the

plunger by means of the flash valve. The motion of the plunger is at all times under the control of the cam, and its movement is therefore in definite relationship with that of the crankshaft.

7. The Adaptability of the Injection Systems to different Classes of Engines.

If the three injection systems are compared, the first two are found to have one important feature in common: the force which governs the flow of fuel through the injector orifice is constant and is independent of the functioning of the engine.

In System (1), since the pressure on the fuel oil remains constant, the quantity of fuel discharged through a given orifice is dependent upon time alone, and in the application of this system to the engine the area and duration of the opening of the flash valve are co-ordinated with the pressure to ensure that the correct quantity of fuel is delivered into the combustion chamber.

In System (2), the force available to accelerate the moving parts of the pump and the fuel within it is provided by the spring. The magnitude of this force is unaffected by engine conditions and depends upon the dimensions of the spring; these are chosen so that in conjunction with the masses of the moving parts and the resistance of the nozzle they ensure that the correct quantity of fuel is delivered to the engine in the required period of time. Here again, the driving force being constant, the quantity of fuel delivered is a function of time alone.

How will this condition influence engine operation? Under the constant-speed conditions of engines in Class I the various factors which govern the ultimate discharge of fuel can be adjusted to give the desired quantity of fuel and the correct rate of delivery to give the optimum results. With System (1) either the period of opening or the lift of the flash valve, or both, can be adjusted when loads other than the maximum are required, and it is possible to regulate matters so that the optimum results can be obtained under all conditions of loading. With System (2) the spring force and pump masses are so balanced that the rate of delivery is such that the optimum results are obtained from the engine under full-load conditions. At other loads the flash valve or spill valve curtails the injection in accordance with requirements. Hence, if the conditions have been correct for full load they will be equally so for part loads, especially as it is simple to arrange matters so that any adjustment in the timing of the commencement of injection is made automatically as the load varies. In theory, therefore, both Systems (1) and (2) are admirably suited to engines in Class I.

Marine engines, Class II, run for by far the greatest part of their time under a constant load; this, for reasons of economy, is usually somewhat less than full load, and they are seldom called upon either

for prolonged running at loads materially different from the normal or for rapid changes in load. When manœuvring, rapid changes in load and speed are called for, but, except in a few special cases such as tugboats and ferries, these operations represent but a small fraction of the running time of the engine. Such variations in load as do occur entail at the same time a definite and predetermined change in speed (see fig. 114). The injection equipment is therefore adjusted for normal output conditions, and if some departure from the optimum results is occasioned at loads widely different from the normal the amount of running under those conditions is usually so short in duration as to be of no great importance.

With System (1) either the lift of the flash valve or the time it remains opened, or both together, may be used to regulate the load. When the lift of the valve is changed, the absolute rate of injection in lb. fuel/sec. is altered, but the duration of injection measured in degrees of crankshaft rotation remains unchanged. The effect upon the rate of injection in terms of lb./degree of crankshaft rotation will of course depend upon the absolute rate of injection and the new engine speed. When the duration of opening is altered, the absolute rate of injection is unchanged, but the rate of delivery per degree of crankshaft rotation increases, while if both the lift and the angle of opening are changed, both the rate of injection and its duration are altered. It should therefore be possible to adjust matters so that satisfactory results are obtained over the whole working range.

With System (2), unless some adjustment is made to the spring load, a cumbersome and somewhat inaccurate method of adjustment, the angular period during which the fuel under pressure is in communication with the injector nozzle must be altered in order to bring about the required changes in load. The plunger velocity then remains constant, but because the power of the engine varies as $(\text{r.p.m.})^3$, the quantity of fuel required will be a function of $(\text{r.p.m.})^2$, and the injection period and rate consequently decrease as the load and speed decrease. Some idea of the nature of the change is given by Table XXII (p. 230). This, however, is inaccurate in one respect, namely, that the quantity of fuel has been assumed to vary directly with the B.H.P., whereas, of course, it will vary more nearly with the I.H.P. The friction losses decrease as the speed decreases, and the cycle efficiency increases, so that it is difficult to make a correct estimate, but the figures given will serve to indicate the general tendency.

As marine engines are not normally called upon to deliver less than about half their output for more than short periods, there should not be much difficulty in obtaining satisfactory results over this range, and System (2) may be considered as suitable for engines in Class II.

When we come to the engines in Class III, however, the situation is very different. The wide and rapid changes in load and speed of

engines in this class render the first two injection systems totally unsuitable unless some fundamental change is introduced into their make-up. Actually the two systems react differently to a wide change in speed, and must therefore be considered separately.

In System (1) the metering of the fuel is governed by the pressure on the fuel, the area of opening of the nozzle, and the *time in seconds* during which the valve remains open. Under conditions of constant torque and varying speed the fuel required per cycle remains constant, but as the engine speed increases the *time in seconds* during which the flash valve remains open decreases proportionately and the quantity of fuel delivered per cycle will therefore decrease in like proportion. The period of injection measured in crankshaft degrees remains unchanged, but unless some change is made either in the pressure or the

TABLE XXII

R.P.M. Per cent of full speed	B.H.P. Per cent of full load	Torque per cent of full load	Crank angle during which pump plunger travels full stroke	Angle required to inject fuel for torque developed
100	100	100	30 deg.*	30 deg.*
90	73	81	32	26
80	51	64	38	24
70	34	49	43	21
60	22	36	50	18
50	12	25	60	15
40	6	16	75	12

area of opening of the nozzle valve the quantity of fuel delivered per cycle will vary inversely as the engine speed, or in other words the quantity of fuel delivered per hour or per minute remains constant irrespective of the engine speed and the indicated power developed will be constant at all speeds. To adjust the pressure and/or the area of the orifice so that the correct quantity of fuel is delivered at all speeds presents considerable difficulties. The angle of opening can, of course, be varied also, but this is undesirable for thermodynamic reasons.

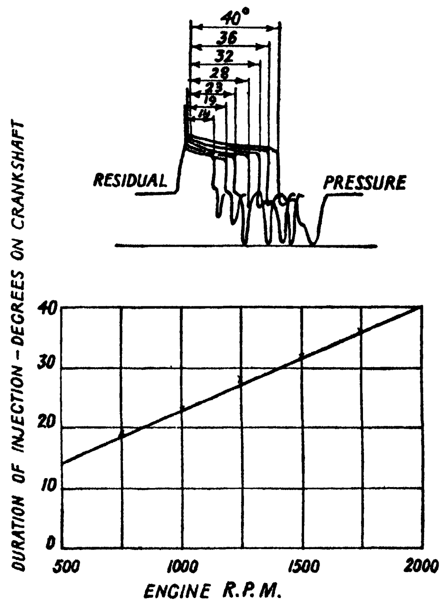
System (2) depends upon a constant force provided by some form of spring, and this force remains unchanged by a change in engine speed, as do also the masses of the pump system. The acceleration of the pump mechanism is therefore independent of the engine speed, so that the *time in seconds* taken by the pump plunger to complete its working stroke is constant and remains unchanged by a change in engine speed. Any change in engine speed is thus accompanied by a corresponding change in the number of degrees through which the crankshaft will rotate during the working stroke of the pump, i.e.

* Angles to the nearest degree.

during the injection period. This increase in injection period is shown in fig. 115, which gives the duration of injection at different speeds from a pump operated solely by spring pressure.

In obtaining these results the only change made was the dynamometer adjustment necessary to allow the change in speed to take place. In the diagram the duration of injection will be seen to vary linearly, although it does not increase quite in direct proportion with the speed. This may be explained by the very much greater gas pressures which occurred at the lower speeds. These constituted an added resistance, and so provided some small measure of automatic compensation tending to prolong the injection.

High gas pressures at the lower speeds are the natural outcome of the greatly increased rates of injection which such an injection system gives at the lower speeds. With this system in its simple form, i.e. without some form of automatic compensation to maintain the injection period at a substantially constant number of degrees of crankshaft rotation, it is quite impossible to obtain a compromise that will give satisfactory results over more than a narrow range of speeds. In the instances quoted conditions were reasonably satisfactory over a speed range from about 1250 to 1500 r.p.m., whereas the required speed range for vehicle work is at least six times this amount. Outside this range, either the maximum pressure reached an excessive value or the combustion became very irregular.



By courtesy of The Automobile Engineer.

Fig. 115.—Increase in injection period with speed with a spring injection system

It is thus seen that although Systems (1) and (2) can be made to fulfil the requirements of engines in Classes I and II, they have very decided disadvantages when the engine is called upon to operate over a wide speed-load range. The disadvantages are sufficient to rule them out completely in their simple forms, and considerable complication is necessary if the drawbacks are to be successfully overcome.

For a variable-speed engine the ideal system is the one which will deliver the fuel at, and ensure that it is burned at, a rate which increases directly with the speed. This means that the cycle of operation must be reproduced exactly at all engine speeds. Such a system is apparently provided by System (3), at any rate in so far as the delivery of the fuel is concerned, as the burning of the fuel is not the responsibility of the injection system alone. In System (3) we have a plunger which is under the direct control of a cam, while the cam itself is driven positively by the engine. The movement of the plunger, therefore, is directly governed by the behaviour of the engine, and the total quantity of fuel delivered, the duration of injection measured in crankshaft degrees, the time at which injection commences, and the rate at which the fuel is delivered per degree of crankshaft rotation should all remain constant regardless of engine speed. On the face of it, the system should be ideally suited to engines having a wide range of speeds, and for that matter to any other engine also, because by the use of a cam of suitable form any desired rate of injection should be obtainable.

While pumps operating on this system do conform in a general way to the ideal just outlined, they provide only an approximation to the ideal, sometimes only a very rough approximation, and in actual practice certain divergences do take place. All the same, with careful design and manufacture, pumps conforming to this system have proved far more successful and adaptable, not only for engines in Class III, but for all three classes, than either of the other systems. It is no exaggeration to say that it was the production of a satisfactory jerk pump that made the high-speed compression-ignition engine possible, and was responsible for the tremendously rapid development of the smaller and lighter classes of engine running at a speed comparable with that of the petrol engine.

Apart from their unsuitability for variable-speed engines, the pressure rail system and the spring injection system suffer from certain other drawbacks which have retarded their adoption in high-speed engines. The presence of a continuous pressure of a very considerable magnitude in System (1) introduces difficulties of its own, the chief of which is leakage. Any leakage at the injector nozzle will not only mean a loss of economy but may also be a source of serious danger to the engine. Any leakage will mean that fuel is introduced at a period in the cycle other than the correct one, and may therefore result in a large quantity of fuel burning at an incorrect point in the cycle and under constant volume conditions, producing very high pressures which may cause serious damage. The movement of the flash valve must be controlled within very fine limits if the delivery of the fuel is to be maintained at the desired rate and quantity, and as the engine speed is increased this becomes an increasingly difficult problem. Although the lift of the flash valve is exceedingly small the masses

involved are of a fairly high order, and the short angular period between the opening and closing of the valve results in a high rate of acceleration. The forces involved are therefore large and make it difficult to retain a proper control over the quantity of fuel injected, owing to the wear that takes place.

The system, however, has its attractions; the idea that the pressure necessary for producing the correct atomization of the charge has already been applied to the fuel, instead of having to be built up when wanted, is a strong point in its favour and has led to repeated, but mostly unsuccessful, projects being put forward with a view to encouraging its adoption. In many of these projects the fact has been overlooked that unless some form of compensation is provided the operating force remains unchanged with speed, and the author has experienced considerable difficulty in getting some of the protagonists of this system to realize this point. That the system is capable of giving excellent results is shown by those obtained by H. B. Taylor at the Royal Aircraft Establishment from the single-cylinder 20.T. experimental engine. This engine was primarily intended for research work on aircraft engines and its speed was limited to a maximum of 1250 r.p.m. These experiments were carried out as long ago as 1926, and a consumption as low as 0·372 lb./B.H.P. hour at a B.M.E.P. of 107 lb./sq. in. and a speed of 1000 r.p.m. was obtained,* while brake mean pressures of the order of 120/sq. in. for a consumption of 0·4 lb./B.H.P. hour were obtained. The dimensions of the engine were fairly large, being 8 in. bore × 11 in. stroke, the engine being intended for airship propulsion. In the multi-cylinder engines subsequently produced for the ill-fated R.101, the system was not used, and a "jerk pump" system was installed.

The spring injection system entails the use of heavy parts, and the high velocity necessary for the freely moving plunger involves the use of a force of considerable magnitude; further, the impact produced in bringing the plunger to rest makes the system noisy. The trip mechanism is prone to wear and tear, which results in irregular action, and the system cannot be considered as suitable for use with really high-speed engines of any kind.

The difficulties associated with both the other systems have resulted in the almost universal adoption of the jerk pump system, not only for high-speed engines, but also for medium- and low-speed engines, and in the case of the latter the manifest advantages of the system have resulted in its displacing the air blast system almost completely.

* H. B. Taylor, High-speed Compression-ignition Engine Research, *Proc. Inst. Automobile Eng.*, Nov., 1927.

8. The Functioning of the Jerk Pump System.

In view of its importance, the "jerk pump" system may well be dealt with in more detail. As most commonly applied, the flash valve, as a separate entity, is dispensed with, and its functions are incorporated with those of the plunger itself. In this case the plunger is provided with a stroke some three times as great as the distance travelled while delivering fuel to the engine, the middle third of the travel being the actual working part of the stroke. The plunger draws in a full charge of fuel during its down stroke, and during the first part of the injection stroke the fuel is spilled back into the supply system through a port drilled in the side of the barrel. At a predetermined point in the stroke the plunger overruns this port and thus traps the remainder of the charge, which is then forced past a delivery valve and is delivered via the injector or "atomizer" into the combustion chamber. When the required quantity has been delivered, the remainder of the fuel in the pump is diverted back to the supply tank either by the plunger uncovering a port or by means of a spill valve being raised from its seat at the desired moment. A valve may be used instead of a port to allow the first part of the charge to escape back to the supply tank, and in this case, for dynamical reasons, separate valves are usually employed for spilling the first and last parts of the charge. The quantity of fuel is regulated by varying the distance the plunger is allowed to travel between the closing of the first port or valve and the opening of the second. This variation may be obtained by varying the distance the pump plunger has travelled either before the first port is closed or before the second port is opened; or both methods may be used together if circumstances suggest that it is desirable. The variation of the distance travelled by the plunger before the first port is closed results in a progressive retardation of the beginning of injection as the quantity of fuel injected is reduced, while the variation of the distance travelled by the plunger before the second port is opened results in an earlier termination of the injection period as the quantity of fuel injected is reduced. By the variation of both the beginning and the end of injection almost any desired combination can be obtained, and it is possible to arrange matters so that, if desired, the injection timing may be advanced as the load is reduced. In this case, however, it must, of course, terminate correspondingly earlier, because with the jerk pump system the rate of delivery per degree of rotation cannot be changed with a change in delivery.

Where valves as such are dispensed with, or if the flash valve, when independent of the plunger, is of the piston-valve type, the positions at which the spill ports are closed and opened again are varied by making either the edge of the plunger which engages with the port, or the edge of the port itself, in the form of a helix. When this method is

used a rotation of the plunger, or of the flash valve, produces the desired variation of the period during which fuel is trapped in the pump barrel and so delivered to the combustion chamber. This method has the great advantage that the effort necessary to rotate the plunger is very small and the regulation of the quantity of fuel is easily and accurately effected.

We do not propose to deal with the various arrangements employed for pumps by different manufacturers, but the general principles underlying the jerk pump system are shown in fig. 116.

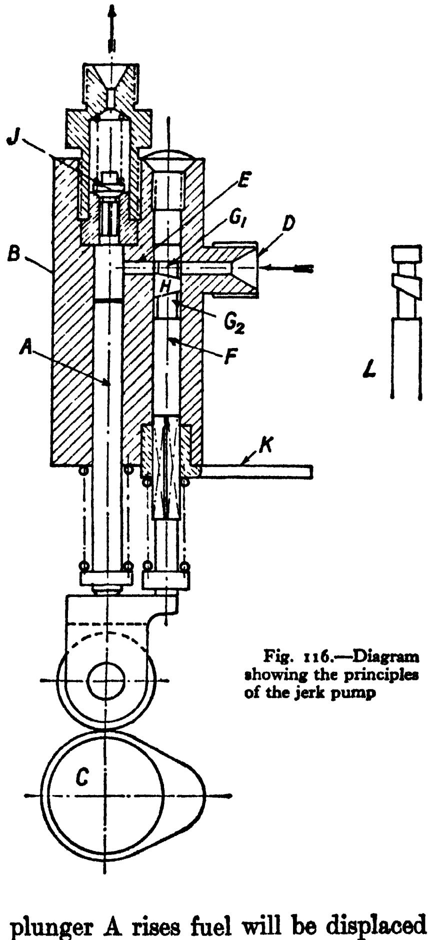


Fig. 116.—Diagram showing the principles of the jerk pump

In this diagram, A represents the plunger working in the pump body B and operated by the cam C. The pump is fed with fuel from the supply pipe D through the drilled passage E. Lying between the supply pipe and the pump chamber is the flash valve F, which may conveniently be operated from the same cam as the plunger. The flash valve consists of a plunger around which are two grooves G_1, G_2, between which lies H, that portion of the plunger which actually functions as the valve. When the plunger A is at the bottom of its stroke, the groove G_1 comes opposite to the passage E and places it in communication with the source of fuel through D. Assuming the whole system to be full of fuel, as the plunger A rises fuel will be displaced from the space above the plunger via E, G_1, and out through D back to the supply tank. The movement of A carries with it the valve F, and at a certain distance up the stroke H overruns the port E; the fuel remaining in the pump is then trapped and forced out through the delivery valve J, and thence via a suitable arrangement of piping and nozzle to the combustion chamber. The continued movement of

A and F ultimately results in bringing the groove G_2 opposite port E, releasing the fuel remaining above the plunger and allowing it to flow back to the supply tank, thereby causing the flow of fuel past J, and therefore the delivery of fuel through the injector, to cease. For the remainder of the stroke the fuel passes out uninterruptedly via E, G_2 and D back to the supply tank. On the return stroke fuel is drawn in as the plunger falls, but is interrupted as H again passes over E; the resulting vacuum is filled up as G_1 comes opposite to E, and when the end of the stroke is reached the pump has been refilled with fuel and is ready to repeat the process.

The quantity of fuel delivered via the delivery valve J depends on the distance travelled by the plunger A while the port E remains closed by H. By making one or both of the edges of H helical in form and providing F with a means, K, by which it can be rotated about its own axis, the distance travelled by A while the passage E is obstructed may be varied at will. The discharge of fuel through J may be entirely prevented by arranging H so that in one position it fails to close E, and the whole of the fuel will then be discharged back to the supply tank. In the diagram both edges of H are shown with a helix, but of opposite hands. This will result in a progressive retardation in the injection timing as the quantity of fuel delivered decreases. By making both helices of the same hand, but of different pitches, as shown at L, the injection timing may be either advanced or retarded as the fuel delivery is reduced. By making either edge of H square with the axis the point of delivery or termination of the injection may be made constant.

By suitably arranging the ports in the body of the pump and adding slots or drilled passages in the plunger the functions of A and F may be combined in a single unit, as is done in many instances with a considerable degree of simplification. It will be realized that by suitable design the helical edge may be incorporated with the port in the pump body if so desired, thereby simplifying the plunger at the expense of a certain amount of added difficulty in making the body.

The foregoing serves to illustrate the general principles of practically all fuel pumps associated with high-speed engines. There are, of course, numerous variations as to detail, the chief of which is the combination of A and F into a single part, which is now almost the standard arrangement. The practice of spilling about one-third of the plunger displacement before starting injection has the great advantage of allowing the plunger velocity to be built up before starting injection. This provides a very rapid closing of the port and gives a sudden building-up of the pressure on the injection side of the pump. This is a very desirable feature, because the steepness of the wave front in the fuel column may have a marked influence upon the result obtained from the engine and, in particular, its freedom from a blue-white colour in the exhaust.

For many purposes it is found that even with this arrangement the steepness of the wave front is insufficient to provide adequate atomization at all times; to overcome this difficulty it is usual to provide the spray nozzle with a spring-loaded differential valve set to open when a given pressure upon the fuel has been produced. The sudden opening of this valve in conjunction with the high pressure on the fuel is then sufficient to produce the desired effect.

On paper such a system appears to be ideal; the cycle of operations should be reproduced with perfection at any and every speed, the required rate of delivery should be obtainable by a suitable cam, and could be varied during the delivery period to any desired extent by shaping the cam accordingly, while the masses, being small, should easily be constrained to follow the cam with exactitude. In addition, the timing could be varied so that the optimum injection point could be obtained at all loads between zero and the maximum. Unfortunately, in actual practice it is only the last two items that can be realized with any great degree of exactitude, and the last one frequently means but little, because in many instances equally good results can be obtained with a fixed timing for all loads at any one speed.

The imperfections of the system, however, lie not so much in the system itself as in the fuel it has to handle.* For all ordinary engineering purposes fluids are looked upon as being incompressible, and for most purposes they may safely be so considered. From the point of view of the injection equipment of a compression-ignition engine, however, the fuel is compressible to an extent that introduces serious difficulties in the way of obtaining a really ideal system of injection. If the fuel were really incompressible in the strict sense of the word, the flow of the fuel at the nozzle would be the exact counterpart of the displacement of fuel by the pump plunger, and we should then be able to control injection in almost any way we desired. This, however, is very far from being the case; the flow of the fuel at the nozzle bears little or no resemblance to that produced by the pump plunger. The column of fuel in the system formed by the pump chamber, fuel pipe, and injector body is to all intents and purposes a stiff spring, and under working conditions behaves as such, the effect being emphasized by an unavoidable lack of rigidity in the pump mechanism and piping. That this fact does not seem to be generally realized is indicated by the tremendous length of fuel pipe seen on some high-speed engines. The effect of the compressibility of the fuel is that pressures generated by the sudden closing of the spill port of the pump are transmitted along the column of fuel in the form of waves, and the purely hydraulic effects in the system are modified out of all recognition by these waves.

* It may be pointed out that the other two systems will also be affected by these imperfections in the fuel.

9. The Effects of the Compressibility of the Fuel.

With a truly incompressible fuel, as soon as the spill port closed, the pressure in the system would rise instantaneously throughout the whole system to the opening pressure of the spray valve. The opening of the spray valve would cause a momentary check in the rise of pressure, owing to the increase in volume as the differential valve was raised to the limit of its travel, but as soon as this limit was reached the pressure would again rise instantaneously up to the pressure required to force the fuel out through the nozzle orifice at a rate equal to that at which it was being displaced by the pump plunger. The reopening of the spill port would produce the reverse effect; the pressure would drop instantaneously to zero, with a momentary check as the nozzle valve closed, and the resulting pressure displacement diagram for the whole process would be somewhat as shown by the full line in fig. 117. A departure from the ideal would be introduced by the fact that the spill port does not close instantaneously, but more or less gradually. Under these circumstances the pressure would build up gradually while the spill port was closing and finish up with an infinite rate of rise to the nozzle valve opening pressure as the port became fully closed. The gradual opening of the spill port would have a similar effect at the beginning of the collapse of the pressure, and these port effects, as they may be termed, would be as shown dotted in fig. 117.

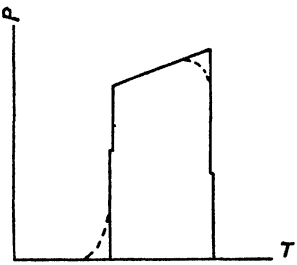


FIG 117—Pressure-time diagram with an incompressible fuel

The compressibility of the fuel introduces a condition differing only in degree from that experienced when dealing with a gas. Instead of an instantaneous pressure rise, the fuel is compressed and satisfies a pressure-volume relationship, as a gas does, but with the distinction that in the case of a liquid the pressure rise is large for a small change in volume. With the closing of the spill valve, therefore, the pressure increases rapidly along the *PV* compression curve for the fuel until the differential valve opens; the opening of the valve causes a momentary check in the rise in pressure accompanied by a small amount of re-expansion of the fuel, followed by a further change in pressure as the pressure rises to the figure necessary to discharge the fuel displaced by the plunger. Exactly what happens during this latter phase depends on the compressibility of the fuel, the resistance offered by the nozzle, and the rate of displacement of fuel from the pump, but with a constant plunger velocity an equilibrium condition will ultimately be reached if the injection is continued long enough.

When the spill port opens a re-expansion of the fuel will take place, the compressed fuel discharging itself through both the spill port and the nozzle orifice. As the differential valve closes, the spring forcing the valve inwards tends to maintain the pressure for a short time and so causes a momentary check in the fall of pressure. As soon as the valve has closed the re-expansion continues uninterrupted to zero pressure, the whole process being as shown by the full-line diagram in fig. 118. The more or less gradual opening and closing of the spill port produce the same effect as described before, and the final results are as indicated by the dotted line in fig. 118.

The necessity for a spring-loaded nozzle valve is very largely due to the combined effects of the compressibility of the fuel and the wire-drawing effect of the gradual opening and closing of the spill port. Combined, they result in a progressive building-up of the pressure at the beginning and a gradual collapse of the pressure at the end of injection, with the result that if an open nozzle is used, there is a period at the beginning and end of injection during which the pressure is insufficient to produce the desired spraying action of the fuel. The effect is very much the same as when a hose-pipe is turned on and turned off again; unless there is a certain minimum pressure drop across the nozzle, a solid stream of liquid is produced instead of the required spray. Such a condition results in inefficient combustion and smoke, and is greatly aggravated at low engine speeds, where the velocity of the pump plunger is low and conditions are therefore not so favourable for the production of the spray. The addition to the nozzle of a spring-loaded valve ensures that a certain minimum pressure is reached before injection starts and that injection ceases when the pressure again falls below a given minimum value, and guarantees a velocity at the nozzle sufficient to ensure adequate pulverization throughout the whole of the injection period.

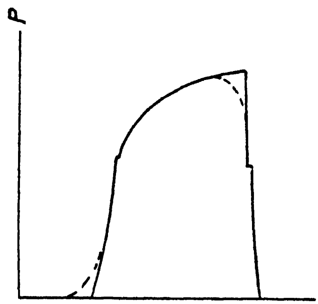


Fig. 118.—Pressure-time diagram with a compressible fuel

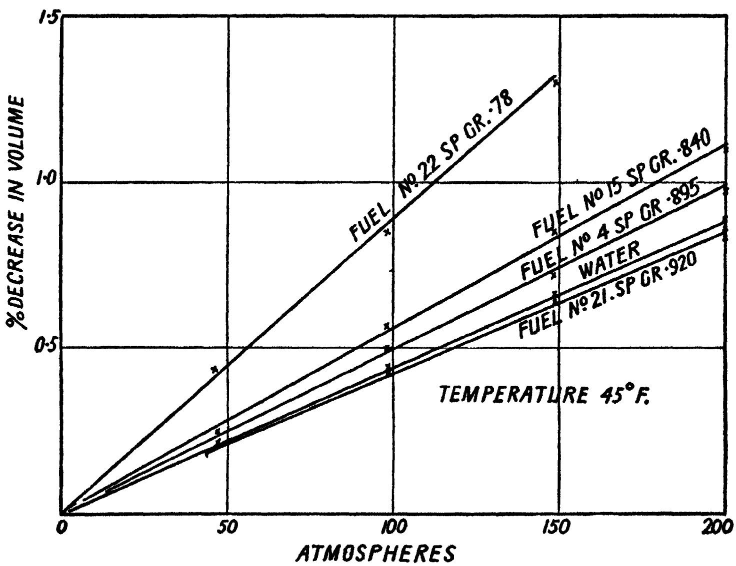
10. The Bulk Modulus of the Fuel.

The compressibility of a fluid is known as the bulk modulus. The bulk modulus is the equivalent of the Young's modulus of a metal, i.e. it is the ratio of stress and strain, the strain in the case of a fluid being cubic and under compression only. The importance of the bulk modulus of the fuel lies in its influence upon the speed with which the pressure waves occasioned by the operation of the fuel pump are transmitted through the liquid. A pressure applied to a body of liquid is trans-

mitted to all points in the liquid with the speed at which sound travels in the liquid. The speed of sound in any liquid is governed solely by the bulk modulus and the density of the liquid, and is given by the equation

$$V = \sqrt{\frac{Kg \times 144}{\rho}} = 8{\cdot}61\sqrt{\frac{K}{d}},$$

where V is the velocity of sound in the liquid in ft./sec., K the bulk modulus in lb./sq. in., ρ the density in lb./c. ft., g has the usual value of 32·2 ft./sec.^{2}, and d is the specific gravity of the fuel.



By courtesy of Diesel Engine Users' Association.

Fig. 119.—Bulk modulus of fuels of differing gravity at different pressures

The bulk modulus of fuel oils varies with the specific gravity, and Table XXIII gives some figures quoted by Le Mesurier and Stansfield.* They state that the bulk modulus is practically independent of pressure up to 200 atmospheres. Some of their results, expressed as a percentage decrease in volume at various pressures, are shown in fig. 119, which is reproduced from the same paper.

According to Le Mesurier and Stansfield, the bulk modulus decreases rapidly with the temperature, and fig. 120, also from the same

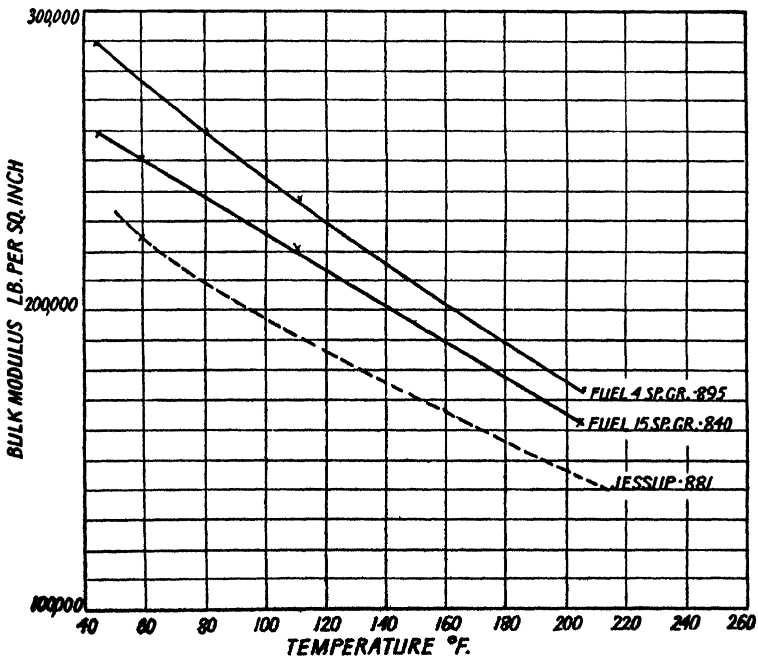
* Fuel Characteristics in Relation to Pump and Sprayer Action, *Diesel Engine Users' Association*, March, 1933.

paper, shows the relationship between the modulus and temperature for three fuels. Referring to this diagram, they say: "It may be

TABLE XXIII

S.G. at 60° F.	Bulk Modulus at 60° F. lb./sq. in.
·895	275,000
·867	262,000
·929	316,000
·978	316,000
·788	206,000
·840	250,000
·859	249,000
·920	310,000
·728	150,000
·881	259,000
·999 (Water)	310,000

taken, at least for these three cases, that the compressibility increases about 55 per cent for a change in temperature from 50° to 200° F.



By courtesy of Diesel Engine Users' Association.

Fig. 120.—Variation in bulk modulus with temperature

For general purposes it may be taken that each 3° F. rise in temperature increases the percentage change in volume by 1 per cent. Thus a fuel changing by 1·0 per cent under a certain pressure at 50° F. will change by 1·01 per cent at 53° F. This change is, of course, not in any way connected with the cubical expansion of a fuel due to temperature change at constant pressure."

The bulk modulus decreases with a decrease in density, so that the effects of a change in modulus upon the velocity of sound tend to cancel out, and as the velocity varies with the square root of, and not directly with, the ratio of the modulus and the density, the variation in velocity with different fuels is not very great. Referring to the effects of temperature upon the velocity of sound in the fuel, Le Mesurier and Stansfield (*loc. cit.*) state that while this is small it cannot always be neglected, and they quote an instance of a fuel in which the velocity was 4850 ft./sec. at 50° F. and 4260 ft./sec. at 150° F.—a difference of 590 ft./sec., or rather more than 12 per cent. They also give a correction factor for the specific gravity of fuel of —0·00035 per °F. rise in temperature as being approximately correct for all compression-ignition engine fuels. As a general figure a velocity of sound of about 4700 ft./sec. at normal temperatures may be considered as an all-round figure for the fuels used with high-speed engines.

11. Wave Effects and the Process of Injection.

At any instant the pressure in the injection system will be governed by the volume of fuel in the system and the decrease in volume caused by the plunger movement, i.e. the increase in pressure is equal to $K \times$ specific decrease in volume. This, however, represents the system under static conditions, and to reach this state a length of time must elapse sufficient to allow the pressure generated by the pump to travel to the most remote part of the system. The equation is, however, equally true when, under dynamic conditions, differences in pressure exist in different parts of the system, if the mean value of the pressure is used. It has already been stated that any pressure applied to one point in a fluid will be transmitted with the speed of sound in the fluid to all points throughout its mass, and there is therefore a time-lag between the application of a pressure at the pump end of the system and its appearance at the nozzle. It might be thought that with a velocity of the order of 4700 ft./sec. the time taken for the pressure wave to reach the far end of the system would be so short as to be immaterial, but at high engine speeds the time required for the wave to travel through a system of normal length represents a by no means inconsiderable period compared with the time available for a complete cycle. In the case of an engine running at 1500 r.p.m., the time taken by the crankshaft to rotate through 1° is $\frac{1}{9000}$ sec., while the time taken for the pressure wave to travel along a fuel

system 2 ft. long is $\frac{1}{2350}$ sec., so that the crankshaft will rotate through nearly 4° while the wave is travelling from the pump to the nozzle. All reactions at the nozzle must therefore lag behind the corresponding action at the pump by nearly 4°. The time required for the wave to travel from the pump to the nozzle is independent of engine speed, so that this lag will vary directly with the speed of the engine and constitute an automatic retardation of the injection timing as the engine speed increases. The significance of this fact will be all the more apparent when it is remembered that a pipe length of 2 ft. is short compared with that found in many high-speed engines to-day.

The fundamental principles governing the fuel injection system have been dealt with in some detail by Drs. S. J. Davies and E. Giffin,* and the following description of the process is based upon this paper.

The pressure of the fuel, instead of rising steadily and uniformly throughout the system, varies along its length, the distance between two points of differing pressure being governed by the time required for the pressure wave to travel between them.

If a fluid is subjected to a change in velocity, a corresponding change in pressure must take place also, the relationship between the two being as follows:

$$\frac{\text{Change in Pressure}}{\text{Change in Velocity}} = \frac{K}{V} = \sqrt{\frac{K\rho}{g}},$$

where K is the bulk modulus, V the velocity of sound in the liquid, and ρ the density of the liquid. The expression holds for either an increase or a decrease in velocity, the sign of the pressure change being appropriate to that of the velocity change.

At the closing of the spill port a velocity is imparted to the fuel along the pipe, which is equal to the velocity of the pump plunger multiplied by the ratio of the areas of the pump plunger and the bore of the fuel pipe, and a pressure corresponding to this change in velocity will be generated at the pump and transmitted along the fuel pipe with the velocity of sound in the fuel. The intensity of this pressure wave is given by the above equation; in actual practice, however, the magnitude of the wave falls somewhat short of the value obtained from the equation, because the closing of the spill port can never be absolutely instantaneous. At the instant when the wave reaches the end of the pipe the pressure in the system is uniform throughout its length and equal to the pressure generated at the propagation of the wave. On reaching the end of the pipe the wave meets with an obstruction either in the form of a sudden constriction from the size of

* An Experimental Investigation of the Flow in Oil Engine Injection Systems, *Proc. I. A. E.*, Vol. XXVII.

the pipe down to the diameter of the orifice, as when an open nozzle is used, or of a closed end, as when some form of spring-loaded valve is fitted to the nozzle. On striking an obstruction the wave will be reflected, partially when the obstruction is only partial, as when an open nozzle is provided, or wholly, as when a spring-loaded valve is used and the intensity of the pressure of the first wave is insufficient to open the valve.

To illustrate the principles involved, Davies and Giffin take the simplest possible case of a system fitted with an open nozzle and discharging to atmosphere, the nozzle having an effective cross-sectional area one-fortieth that of the fuel pipe and the pump plunger having an area twenty-five times that of the cross-section of the pipe. The plunger is assumed to have a uniform velocity of 1 ft./sec., and the time of delivery of fuel at the pump is taken as 0·004 sec. The beginning and end of delivery are assumed to take place instantaneously, and the bulk modulus of the oil is taken as 266,500 lb./sq. in., and the specific gravity as 0·86, giving the speed of sound in the fuel (V_s) as 4800 ft./sec. The length of the pipe is taken as 23 in., so that the time required for the pressure wave to travel from one end of the pipe to the other is $23/(12 \times 4800) = 0{\cdot}0004$ sec., or one-tenth of the delivery period.

All these conditions, except the instantaneous beginning and ending of the injection, are realizable in practice. The closing and opening of the spill port or valve require time, and although the action may take place with exceeding rapidity, it must always fall short of being instantaneous. The shorter the time taken to open or close the port, the more rapid the change in pressure.

The conditions existing in the system just described are determined by Davies and Giffin as follows: "With a pump plunger velocity of 1 ft./sec., the corresponding velocity in the pipe is 25 ft./sec., and the magnitude of the pressure wave required to initiate the velocity in the pipe will be $P = \frac{K}{V_s}$ (increase of velocity in pipe) $= \frac{266{,}500 \times 25}{4800}$ $= 1388$ lb./sq. in. In an actual injection system the wave front will be sloping, depending upon the rate of rise of pressure at the plunger face, as determined by the conditions of the suction valve or port. The assumption of instantaneous beginning of delivery gives a vertical wave front.

"This increase of pressure, 1388 lb./sq. in., is propagated along the pipe with the speed of sound, and thus reaches the nozzle after a time interval of 0·0004 sec. With the open nozzle considered, injection begins immediately, and the reflection of the wave is then only partial —if the end of the pipe were closed reflection would be complete, the pressure being doubled as the wave returned. The difference between complete and partial reflection is, as stated earlier, due to the momen-

tum of the oil flowing on through the nozzle. The intensity of pressure, Pr, of the reflected wave is found as follows:

"The total pressure across the nozzle after reflection is $(1388 + Pr)$, so that the velocity through the nozzle

$$V_n = \sqrt{\frac{2g(1388 + Pr)}{\text{density}}}$$

$$= \sqrt{\frac{2g(1388 + Pr)144}{0{\cdot}86 \times 62{\cdot}5}}$$

$$= 13{\cdot}14\sqrt{(1388 + Pr)}.$$

The corresponding velocity in the pipe, $V_p = \frac{V_n}{40} = 0{\cdot}3285\sqrt{(1388 + Pr)}$

Now, the reflected-wave Pr is proportional to the reduction of velocity in the pipe from 25 ft./sec. to V_p, so that

$$Pr = \frac{K}{V_s} \times (\text{Change in velocity}) = \frac{K}{V_s}(25 - V_p)$$

$$= \frac{266{,}500}{4800}[25 - 0{\cdot}3285\sqrt{(1388 + Pr)}],$$

giving $Pr = 579$ lb./sq. in. This gives the total pressure at the nozzle of $(1388 + 579) = 1967$ lb./sq. in.

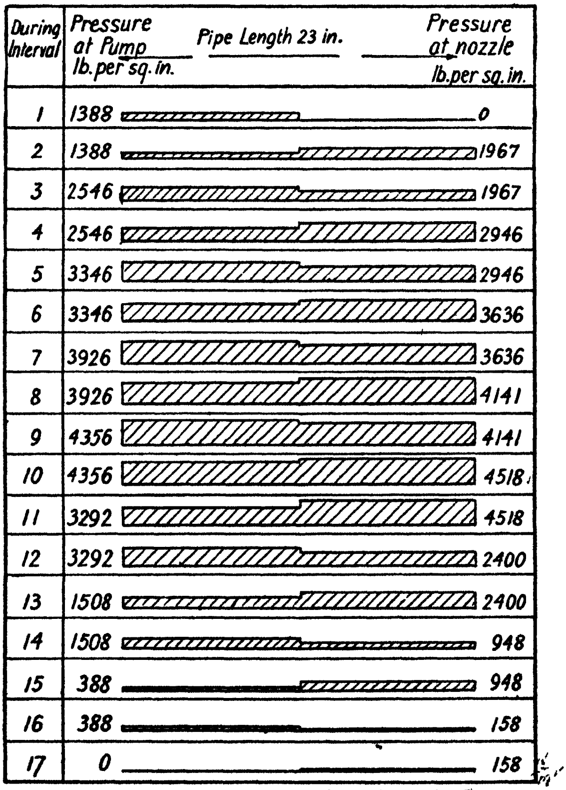
"This reflected wave now travels backwards towards the pump plunger, and the conditions at the nozzle remain constant until this wave, which is completely reflected at the pump, again reaches the nozzle. This complete reflection at the pump is due to the increase in the velocity of the oil at this end of the pipe from V_p to its initial velocity of 25 ft./sec., resulting from the continued steady motion of the pump plunger. The total pressure in the pipe after the passage of this completely reflected wave from the pump to the nozzle is now $1967 + 579$ (or $1388 + 2 \times 579) = 2546$ lb./sq. in. On the arrival of this wave of 579 lb./sq. in. at the nozzle the magnitude of the next partially reflected wave is found by a calculation similar to that given above, and amounts in this case to 400 lb./sq. in.

"The results of the continued calculation over the whole injection period are given in fig. 121.

"The delivery from the pump, it will be remembered, occupies 0·004 sec., that is, ten intervals of 0·0004 sec. At the end of the tenth interval, delivery into the pipe ceases, and the wave reaching the pump end of the pipe at this instant is not reflected. Instead, the oil at this end of the pipe is brought to rest, and a negative wave of 1226 lb./sq. in. brought about by this change of velocity, assumed instantaneous, is now propagated towards the nozzle. When this negative

wave reaches the nozzle it is partially reflected, as before, still further reducing the pressure in the pipe. This process continues until the pressure at the nozzle and in the pipe falls to that of the atmosphere, which, in the case considered, occupies a further seven intervals.

"It is now possible from the data in fig. 121 to compare the process of injection with the rate of delivery from the pump. Fig. 122 thus



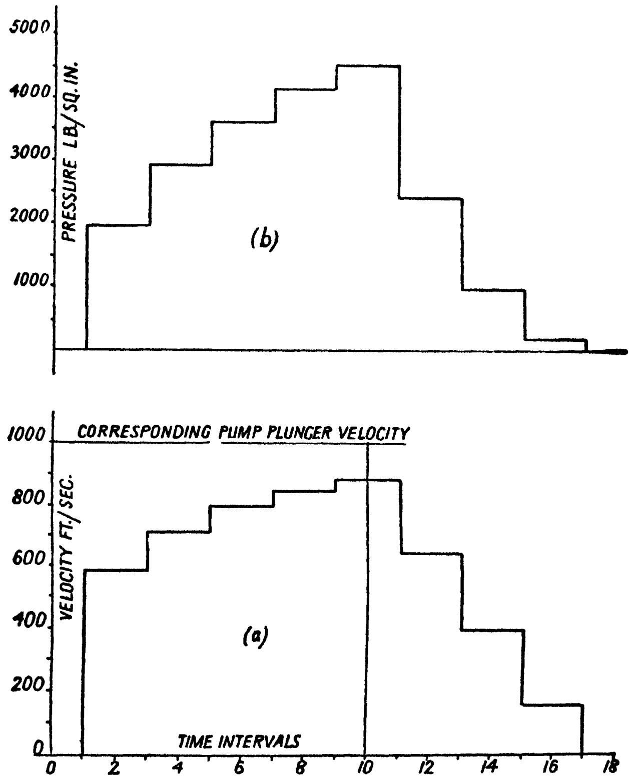
By courtesy of Inst. of Automobile Engineers.

Fig. 121.—Pressure on fuel pipe during injection

shows, on a base of time, comparative curves of rate of injection and rate of delivery from the pump for the case considered, given respectively as actual nozzle velocity V_n and 'corresponding pump-plunger velocity' (= Actual × 25 × 40). The first point to be noticed is that injection at the nozzle does not begin until the end of the first interval, corresponding to the time taken by the initial pressure-wave front to travel from the pump to the nozzle. This is termed the 'injection lag', and in the simple system considered depends for a given fuel and

pipe material on the length of the pipe only, being independent of engine speed and pump-plunger velocity.

"It will be seen in fig. 122 (*a*) that the nozzle velocity or rate of injection increases in steps. These steps are of decreasing magnitude, the velocity tending towards that corresponding to the displacement of the pump plunger. In the present case, in which delivery from the



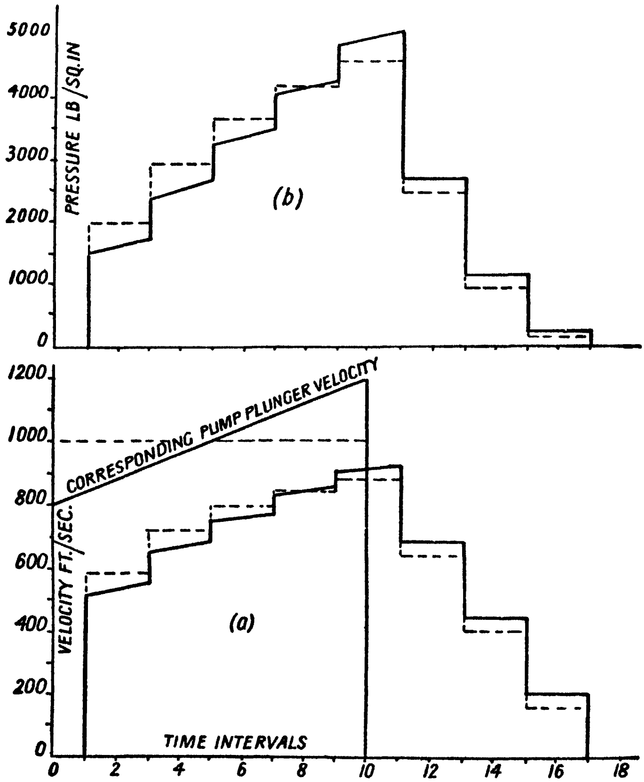
By courtesy of Inst. of Automobile Engineers.

Fig. 122.—Variation of velocity and pressure at nozzle during injection. Plunger velocity constant

pump ceases at the end of the tenth interval, the nozzle velocity at the end of the eleventh interval—allowing for the lag—is 884 ft./sec. instead of 1000 ft./sec., which would correspond to the pump-plunger displacement. Discharge from the nozzle continues, but with a rate decreasing in steps as shown in the figure. Thus, when the time of pump delivery is 0·004 sec., injection continues during 0·0064 sec. So that, even in the simplest case, as a result of the manner in which the pressure is built up in the pipe and afterwards released, the conditions at the nozzle are always different from those at the pump, the

maximum rate of injection being less, and the time of injection longer, than the values corresponding to the motion given to the plunger."

The presence of a pressure greater than that of the atmosphere at the nozzle, which occurs under actual working conditions, modifies matters somewhat, because the pressure difference across the nozzle



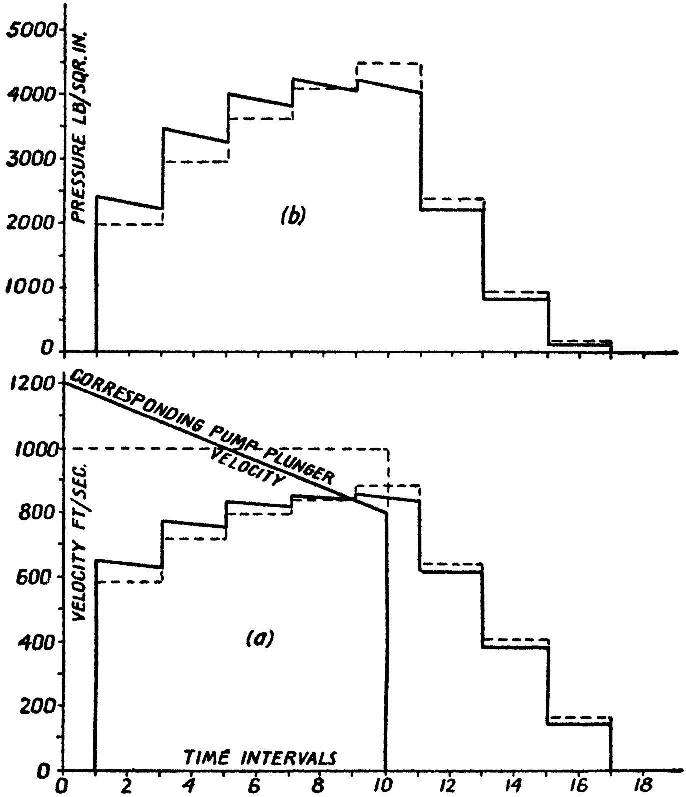
By courtesy of Inst. of Automobile Engineers.

Fig. 123.—Variation of velocity and pressure at nozzle during injection. Plunger velocity increasing

is reduced to the extent of the back pressure, the velocity through the nozzle is correspondingly reduced, and the velocity in the pipe, V_p, is reduced in proportion.* The back pressure produced by the cylinder gases, of course, has not a fixed value, but increases as the compression pressure increases, to be followed by a much more rapid rise in the pressure as combustion takes place.

* In the simple system just described, where no non-return valve is fitted at the nozzle, any pressure on the outside of the nozzle will, of course, be transmitted to the fuel within, and the process of injection will start with the fuel already under pressure to that extent.

The effects of a variation in plunger velocity are investigated by Davies and Giffin on the assumption of a mean velocity equal to that of the constant velocity cam, and it is shown that with an increasing plunger velocity the velocity of the fuel at the jet starts with a somewhat lower value but finishes with a slightly higher one, whereas with

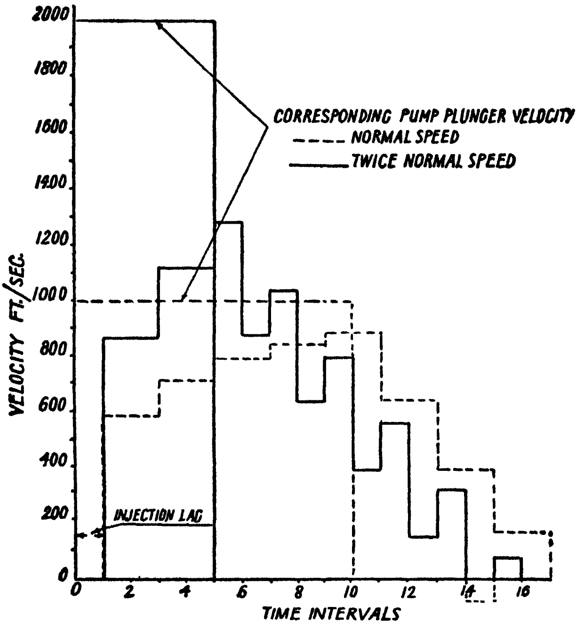


By courtesy of Inst. of Automobile Engineers.

Fig. 124.—Variation of velocity and pressure at nozzle during injection. Plunger velocity decreasing

a decreasing velocity of the plunger the velocity at the jet starts at a somewhat higher value and finishes with a lower one than in the case when a constant plunger velocity is used. When a decreasing plunger velocity is used, the delivery does not begin at a high value and fall to some lower value as the injection proceeds, but builds up during the process as it does when the constant velocity cam is used, but at some lower rate. Actually, the rate of delivery of the fuel tends to be more nearly constant with a falling rate of plunger velocity. Figs. 123 and 124 show the results obtained by Davies and Giffin.

The effect of a change in engine speed of two to one is to double the injection lag in degrees, but measured in time the lag is constant. The delivery time is halved and the plunger speed doubled. The increase in velocity in the pipe at the first forward wave is doubled and the intensity of this wave is therefore doubled. The velocity through the nozzle, however, is not doubled, for it depends on the square root of the pressure. In the example quoted from Davies and Giffin the maximum velocity is increased only from 884 ft./sec. at the



By courtesy of Inst. of Automobile Engineers.

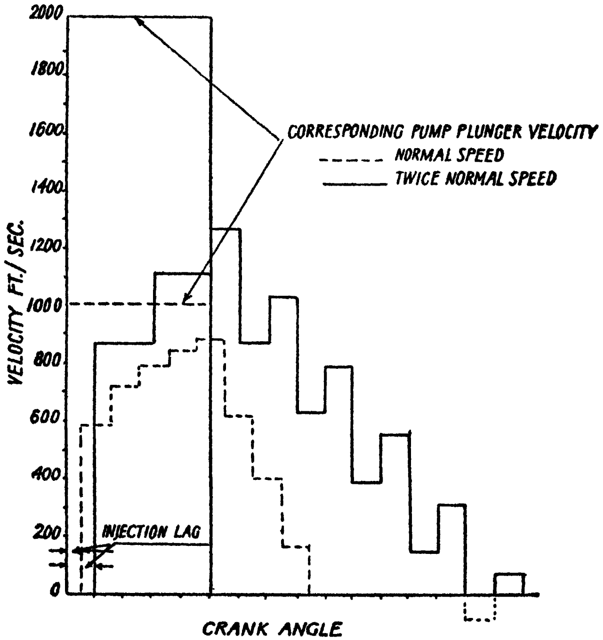
Fig. 125.—Variation of velocity at nozzle during injection: effect of engine speed

normal speed to 1275 ft./sec. for twice the speed, and the actual time of delivery is almost as long as for the lower speed, as will be seen from fig. 125, which is reproduced from their paper. Compared on a basis of crank angles, the pump delivery angles are nearly equal, but at the higher speed the injection period is nearly doubled because of the much longer period required for the pressure to fall to zero after the termination of the pump delivery, as is shown in fig. 126. This is a matter of considerable importance.

The form of the curve during the collapse of pressure at the higher speed is somewhat different from that at the lower speed. Under the conditions considered in the first place the initiation of the negative wave at the plunger coincides with the arrival of a positive reflected

wave from the nozzle. At the higher speed, the negative wave leaves the plunger end at the same instant as the reflected wave leaves the nozzle, and the conditions at the nozzle change at half the time interval; the action between the alternate positive and negative waves produces the curve shown.

One effect of an increase in speed is to increase the rate of closing of the spill valve, or port, and so to produce a closer approximation to an instantaneous closing. The front of the wave therefore approaches



By courtesy of Inst. of Automobile Engineers.

Fig. 126.—Variation of velocity at nozzle during injection: effect of engine speed

more nearly to the vertical, a matter of some importance when a closed nozzle is used. At release, the spill port opens more quickly also.

The effect of a change in fuel pipe length is to change the time interval required for the wave to travel the length of the pipe, the time interval varying in proportion to the length of the pipe; with a shorter pipe, therefore, the successive reflections recur at shorter intervals of time. The increase in pressure at any given reflection, e.g. the fourth, is dependent solely upon the change in velocity at that reflection and, other things being equal, will be independent of pipe length. The pressure will consequently build up more rapidly with a shorter pipe than with a longer one, and for the same reason the fall in pressure at the end of the pump delivery will be more rapid

with a shorter pipe, but in this case the interaction between positive waves from the nozzle and negative waves from the pump may cause a modulation of the shape of the falling pressure curve similar to those seen to occur with the change in speed.

12. The Practical Injection System.

The practical injection system is usually somewhat more complicated than that just described, and is provided in all cases with at least one valve in the form of a non-return valve at the pump, and in the vast majority of cases with a valve at the nozzle also. The necessity for at least one valve on the delivery side of the pump if the system is to function at all will be obvious. The need for the second is not quite so obvious, but for the moment it will suffice to say that this valve is fitted for the purpose of improving the spraying capabilities of the nozzle. The use of a lightly loaded valve at the nozzle to act as a gas check does not materially influence the system, and the presence of such a valve does not constitute a real departure from the open nozzle.

As we shall see later, the valve at the pump may take one of many different forms, and may be either spring-loaded or otherwise, as desired. Usually it is provided with at least a light spring so as to ensure that it remains on its seat unless a delivery of fuel is actually taking place, but it is not necessarily spring-loaded in the usually accepted sense of the word.

The valve at the nozzle, when such is fitted other than as a simple gas check, is usually more or less heavily loaded and may open either outwards towards the combustion chamber or inwards towards the injection system. The latter is by far the commoner, and in actual fact the outward-opening valve has rarely proved successful. The inward-opening valve is of the differential type, the fuel pressure acting against an annulus around the valve proper to lift it from its seat. The pressure required to lift the valve depends upon individual requirements, but a lower limit is imposed by the necessity for having sufficient load on the valve to ensure that it will close promptly against the gas pressure. At the end of injection the gas pressure will be acting against the area represented by the major diameter of the annulus and may amount to anything up to 70 atmospheres. The opening pressure must therefore at least be equivalent to this pressure multiplied by the ratio of the areas exposed to the fuel pressure before and after the valve has opened, and in actual practice this pressure will be found to be not much below 100 atmospheres.

With this arrangement the conditions at the end of the pump delivery will be radically different from those of the simple open system previously considered. The nozzle valve is now loaded so that it cannot open until the pressure on the fuel reaches a predetermined figure,

so that if the first wave to arrive is of insufficient magnitude to open the valve it will be totally reflected and the pressure at the valve will be doubled. This doubling of the pressure may then be sufficient to open the valve and to cause a discharge of fuel to take place. This discharge of fuel will cause a drop in pressure, which may be sufficient to cause the valve to close again. In these circumstances the valve will remain closed until the returning wave again builds up the pressure sufficiently for the valve to open. Under certain conditions the fall in pressure following upon the re-opening of the valve may again be sufficient to cause the valve to close, and this may happen a third time and even more, so that the injection takes the form of a series of short periods instead of a single prolonged one.

Should the first wave be sufficiently powerful, the valve may be lifted at the first impact and injection may begin immediately and continue without interruption, as when an open nozzle is used. The moment of the opening of the nozzle valve is thus governed by the intensity of the pressure wave, the loading on the nozzle valve, and the area exposed to the fuel pressure. The loading on the valve is, of course, governed by the strength of the spring applied to the valve. With a closed nozzle, therefore, the injection lag is increased by the time required for the fuel to reach the opening pressure of the nozzle and, except in cases when the valve is opened by the first wave to arrive, will be longer than in the case of an open nozzle.

The opening of the nozzle valve will in all cases cause a momentary drop in pressure. This will be accentuated with a differential valve by the increase in volume created by the inward movement of the valve. Even when this fall in pressure is insufficient to cause the reclosing of the nozzle, it is frequently sufficient to cause a partial closing, and a pause in the movement of the valve is very commonly met with. The area of the valve exposed to the pressure of the fuel is increased by the opening of the valve, and this increased area assists the fuel in driving the valve to its maximum lift. This increase in area also results in the valve, once opened, remaining open until the pressure has fallen materially below that at which it opens, the difference between the two pressures being in the same ratio as the areas exposed to pressure when the valve is open and when it is closed.

The presence of a valve at the nozzle may modify the system considerably. With an open nozzle the only variable is the change in gas pressure during injection, but with a closed nozzle the behaviour of the valve itself introduces modifications.

The behaviour of the valve after first leaving its seat will depend upon a number of factors: the momentary drop in pressure which follows the opening of the valve, the increase in area exposed to the fuel pressure, the rate of the spring, the masses of the dynamic system, and lastly, the resistance to flow offered by the nozzle and the situation

of this resistance; these latter also govern the drop in pressure which follows the opening of the valve.

Nozzles provided with a spring-loaded valve may be placed in one of two categories: (1) those in which the resistance to the flow of the fuel is mainly due to the opening between the valve and its seat, and which may therefore be termed "variable-orifice" nozzles, and (2) those in which the resistance is provided by the orifice on the downstream side of the nozzle, and which may therefore be termed "fixed-orifice" nozzles. This latter category will, of course, include all nozzles of the open type.

The discharge characteristics will be subject to considerable modification by the movements of the valve; this is particularly true of nozzles of the variable-orifice type, because any oscillation of the valve causes a corresponding alteration in the area of the orifice and therefore in the main source of the resistance of the nozzle. This alteration causes a variation in the change in velocity at the nozzle, with corresponding changes in pressure. Even when fed from a steady source of pressure, spring-loaded valves show a marked tendency to oscillate unless they are given a definite stop to limit their lift and the pressure is sufficient to raise the valve to the limit of its travel and hold it there. Under these conditions the orifice virtually becomes a fixed-orifice nozzle, but at all rates of discharge less than some critical value oscillation will take place. The reason for this oscillation is an instability caused by the reactions between the pressure drop across the opening and the variation in the area of opening; the valve, on opening, is carried by its momentum to a height above that which is necessary for the available pressure to discharge the fuel at the rate at which it is being delivered to the orifice, and the pressure therefore falls and allows the valve to close slightly. The momentum of the valve causes it again to overshoot the correct position, so that the pressure rises and again increases the lift, and the valve will continue rising and falling at a rate governed by the characteristics of the spring and the masses of the system, i.e. with a period of vibration equal to the natural frequency of the system. The instantaneous discharge of the nozzle will be governed by the area of the opening and the pressure drop across the opening, and is therefore governed by the oscillation of the valve. Under such conditions the rate of delivery will fluctuate over a wide range, and in extreme cases the injection may be divided into a number of short injections, as has already been described.

Fixed-orifice nozzles are not so subject to oscillation, although it may take place under certain conditions. The drilled orifice constituting the major resistance to the flow has a drop in pressure across it greater than the drop in pressure across the valve, and the pressure against the exposed face of the valve is usually sufficient to raise the

valve to a position where the area of the passage through the valve is greater than the area through the orifice. Fluctuations in the valve lift, even when they occur, cannot, therefore, influence the delivery to anything like the same extent as in the case of the variable-orifice nozzle. The variations in pressure caused are of a relatively minor nature, and as the discharge is proportional to the square root of the

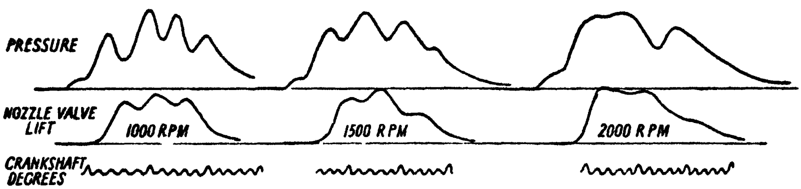


Fig. 127.—Fuel pressure fluctuation and nozzle-valve lift· pintle nozzle

pressure drop across the orifice the fluctuation in delivery is of a much smaller order than in the previous case. At a very low rate of delivery it is possible for the opening through the valve to become the major resistance; the rate of delivery will then be governed by the oscillations of the valve, and the nozzle will virtually have become a variable-orifice nozzle. This condition, however, usually takes place only at speeds considerably below those at which the nozzle is normally intended to operate.

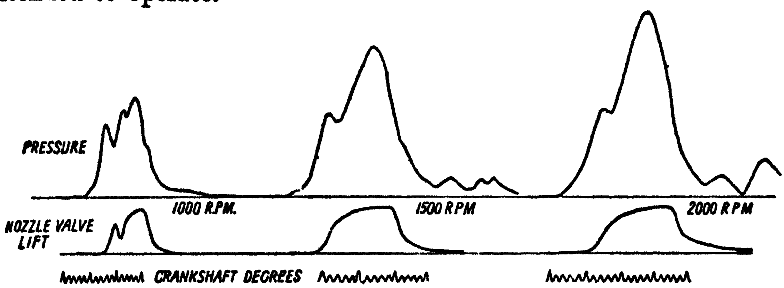


Fig. 128.—Fuel pressure fluctuation and nozzle-valve lift: multi-hole nozzle

It will thus be seen that when a closed nozzle is used the type of nozzle will exercise a considerable influence over the precise rate and nature of the fuel delivery to the engine, and that under certain conditions the fixed-orifice nozzle may function as a variable-orifice nozzle and vice versa.

The oscillation of the valve of a variable-orifice nozzle is well shown in fig. 127, which gives both the variation in nozzle-valve lift and also the fluctuations in pressure at the nozzle at several speeds. The steady movement of the fixed-orifice nozzle is shown in fig. 128, which again shows both the nozzle-valve lift and the pressure at the nozzle.

In fig. 127 it will be noted that the number of oscillations tends to decrease as the engine speed increases. This is because the time period of oscillation is independent of the speed of the engine, but depends only upon the dynamics of the nozzle system. In fig. 128 the partial functioning of the fixed-orifice nozzle as a variable-orifice nozzle is seen at the lower speeds, when the nozzle valve hardly lifts from its seat at the beginning of injection, but later rises normally and reaches its maximum lift.

At the end of the delivery of fuel from the pump the spill port opens and generates the negative wave which travels along towards the nozzle; this latter constitutes an obstruction, and some reflection will occur there, while part of the wave is transmitted past the valve. The movement of the fuel towards the pump will tend to close the pump delivery valve, and once this valve has dropped upon its seat no further reduction in pressure can occur from any action at the pump end of the pipe. Meanwhile the discharge of fuel through the nozzle has continued, causing a fall in pressure at the nozzle end, and even without any assistance from the negative wave from the pump end the pressure will ultimately fall to the value at which the nozzle valve will close. The valve will snap shut and a pressure will be retained in the pipe equivalent to the mean pressure in the system at the moment the second valve closes. In the extreme case the delivery valve closes instantly when the negative wave reaches it and the pressure retained will be equal to the closing pressure of the nozzle. Under more favourable conditions the negative waves may have time to reduce the pressure to a figure materially below that at the closing point of the nozzle valve, and in such circumstances not only will the residual pressure in the system be reduced, but the termination of the delivery of fuel through the nozzle will be more abrupt and thus will tend to avoid any semblance of fading away, such as would be experienced when the previous condition exists.

The presence of a valve at the nozzle terminates the injection much earlier than when an open nozzle is used because, instead of the delivery of fuel continuing until the pressure on the fuel falls below that of the gas pressure against which it is being delivered, the nozzle valve closes as soon as the fuel pressure has fallen below the minimum pressure necessary to keep it open, and so terminates the injection. It thus materially reduces the extent to which the injection is prolonged as the result of an increase in engine speed, as was illustrated in fig. 126.

With a closed nozzle the actual timing of the injection will be varied somewhat by the presence of any pressure remaining in the system from the previous injection, because the increase in pressure to be generated by the pump before the nozzle valve opens will be reduced to the extent of the pressure left in the system, and the injection lag will therefore be reduced accordingly.

13. The Open Nozzle versus the Closed Nozzle.

The question of the relative merits of the open nozzle and the closed nozzle is an important one. The almost primitive simplicity of the open nozzle should prove a very strong incentive to its adoption; the only part of an open nozzle needing any special accuracy is the orifice itself, the diameter and length of which must be made to close tolerances. For the rest, a simple form of non-return valve to prevent gas from getting back into the system is all that is required. Against this, the closed nozzle requires a differential valve and seating made to exceedingly close tolerances. The valve and seating require to be made in specially hardened material, and the position of the seating is usually such that its accurate manufacture introduces a number of problems. The spring for loading the valve requires special attention; it requires to have a high rate of loading, which must be accurate, and the necessary load must be retained under conditions of prolonged loading. The parts connecting the spring and the valve require special materials in order to avoid any indentation under the high unit loading produced at the points of load transmission. The presence of the valve with its spring and the necessary means for adjustment make a far more bulky arrangement than is necessary with an open nozzle, and on small engines the space required for the complete unit is frequently a matter of importance. The effects of temperature upon the closely fitting working parts of the valve may also cause trouble. On the score of simplicity, cheapness, convenience, and compactness there is thus every reason to favour the open type of nozzle, but in actual fact in the vast majority of cases it is the closed type that is employed.

The reason for this lies mainly in the lack of energy in the initial pressure wave. When the open nozzle is used, this lack of energy causes the first portion of the fuel charge to issue from the nozzle with insufficient velocity to produce the requisite degree of fineness of atomization; this condition becomes particularly marked at low engine speeds, and the coarseness of the spray causes a decrease in combustion efficiency and a blue smoke in the exhaust. This defect may be overcome by the use of an air swirl of exceptional vigour, as is done in the Ricardo " Vortex " engine, but in most instances the production of the necessary swirl either decreases the mechanical efficiency of the engine to an extent which entails too great a loss or reduces the breathing capacity of the engine to an undesirable extent. The problem becomes one of increasing difficulty when the engine is required to operate over a wide speed range and may also be required to idle at a speed considerably below its normal operating r.p.m. Under such conditions both the vigour of the swirl and the energy of the pressure wave are greatly reduced, with the result that the small

quantity of fuel then required is apt to arrive in a few globules of relatively large size. The imperfect combustion that follows causes an accumulation of partly burned fuel in the engine cylinder and exhaust passages, and when the engine is again placed on load a cloud of bluish-white smoke is produced and may persist for an appreciable period. A further trouble is that these products of partial combustion are apt to cause sticking of the piston rings. This condition is obviated by the provision of a spring-loaded nozzle valve. By its use, injection is delayed until a predetermined pressure has been reached, and the sudden application of this pressure across the orifice produced by the opening of the valve ensures a definite degree of atomization at all engine speeds, and so enables the combustion efficiency to be maintained without the necessity for resorting to an air swirl of abnormal vigour.

A further advantage is that at the termination of injection the valve snaps shut as soon as the pressure falls below the closing pressure of the valve, thus preventing a gradual reduction to zero of the pressure across the orifice and a low rate of injection with a loss of atomizing ability at the finish. The duration of injection is reduced by the cutting off of the toe seen in the diagrams in figs. 122 to 126.

Although the presence of the valve results in an addition to the injection delay, this is not any real disadvantage and can be compensated for by a slightly earlier closing of the spill port, and under favourable conditions the presence of the valve may provide some measure of compensation for the normal increase in injection delay caused by an increase in speed. With the open nozzle, since injection starts as soon as the first pressure wave reaches the nozzle, the injection timing must automatically become later when an increase in speed takes place, but with a closed nozzle it is possible for the injection timing to be *advanced* with an increase in speed. The increase in speed results in a more powerful wave being sent along the pipe towards the nozzle, and as a result the rate at which the pressure builds up before the nozzle valve opens will be increased. If, therefore, the nozzle valve does not open at the impact of the first wave it will be opened earlier, in time, if not in crankshaft degrees, by the more powerful wave produced by an increase in speed. That this is not an unimportant point will be illustrated by the readings given in Table XXIV, which gives the injection timing, as determined by the lifting of the nozzle valve, over a range of speeds for a certain engine.

TABLE XXIV

Actual r.p.m.	756	1023	1252	1504	1748	1992
Injection Timing (deg. before T.D.C.) ..	15	14	14	16	14	15

Such a measure of automatic compensation is a valuable feature.

The provision of a spring-loaded valve is not without its disadvantages, however, and in actual fact it introduces some very definite disadvantages from which the open nozzle is free. The presence of the valve places a limit to the minimum quantity of fuel that can be injected. This minimum quantity will depend upon the pressure at which the nozzle valve is set to open, the bulk modulus of the fuel, the quantity of fuel in the system, and the mechanical springiness in the system. A definite quantity of fuel must first be pumped into the system in order to produce the pressure necessary to open the nozzle valve, and if the quantity delivered by the pump does not exceed this figure no injection can take place. If this minimum quantity is just exceeded by a bare margin an injection will take place, but as the pressure will begin to fall immediately the valve is opened, the valve will close again as soon as the pressure drops to the closing pressure of the valve, and the quantity of fuel delivered will be small. If the valve has failed to open, a part of the fuel may be retained in the system between the delivery valve of the pump and the nozzle valve, and at the next stroke of the pump the quantity of fuel delivered may then be sufficient to bring the pressure up to the opening pressure of the valve and a double injection will take place. This process is commonly termed " eight stroking ", and quite often occurs at light loads, especially at low speeds, such as when idling, and is the cause of irregular running under such conditions. It is a particular source of annoyance with vehicle engines, producing erratic idling, since the relatively light flywheels of such engines store up insufficient energy to smooth out the resulting wide variations in speed. The use of a high opening pressure increases this difficulty considerably.

14. Residual Pressure.

The most serious objection to the use of a closed nozzle is a tendency for pressure to be locked up between the nozzle valve and the pump delivery valve. This pressure is not in itself objectionable, and may indeed be advantageous in some ways, but it causes two troubles. In the first place, the amount of pressure retained is apt to vary with the load and/or speed of the engine, so that the injection timing varies owing to the time required to bring the pressure up to the opening pressure of the nozzle changing with the initial pressure in the system. The injection timing may be altered by a number of degrees by variations in the residual pressure, and both proper functioning of the engine and control over the maximum pressure will be impaired. The second trouble is the more serious of the two, and is perhaps not, strictly speaking, due to the residual pressure. This is a tendency to cause dribbling at the nozzle. The dribble which causes real trouble is not, as is frequently supposed, an actual dripping of fuel from the nozzle as water drips from a leaky tap, but a very slow creeping of

fuel from the nozzle, which hardly does more than cause a wetness around the orifice. Under extreme conditions, as when the pressure across the nozzle falls away slowly instead of finishing abruptly, actual dripping may occur, but this is not nearly so common as the slow seepage. Any fuel around the outside surface of the nozzle is decomposed by the heat and leaves a carbon deposit which builds up around the orifice, and also within the orifice itself, and ultimately interferes with the spraying of the nozzle and the distribution of the fuel in the combustion chamber.

To obviate this difficulty either the nozzle valve must be maintained absolutely tight at all times and under all conditions, or some means must be adopted for avoiding the retention of pressure in the system. The former is, of course, the proper course to adopt whether pressure is retained in the system or not, but under the high temperatures at which the nozzle is called upon to operate it seems impossible to ensure that it will remain absolutely tight against pressures of any real magnitude. Even the most minute leakage will cause trouble. One solution is to make provision for the adequate cooling of the nozzle, and if the nozzle temperature can be kept low enough the cracking of the fuel will be prevented and trouble therefrom will be avoided. This is, however, a very difficult matter in the case of small high-speed engines, where the accommodation of even a small injection equipment is not always easy.

To omit the delivery valve from the pump and allow the nozzle valve to perform the functions of the delivery valve might be felt to meet the situation, but by actual experience the author has found that with the pump delivery valve in its place the engine will operate satisfactorily with a nozzle valve that has a lack of tightness sufficient to prevent the engine from operating at all when the pump valve is removed. Without the pump valve, gas can enter the system and the engine may refuse to start at all or fail to pick up again after the fuel has been cut off for a few revolutions. Apart from this, the pump delivery valve plays an important part in the speed-delivery characteristics of the pump. A delivery valve at the fuel pump is therefore essential, and other means must be adopted if pressure in the system is to be prevented and trouble is to be avoided.

A number of devices have been proposed for relieving the pressure at the end of injection, e.g. opening the system momentarily to atmosphere. The most successful and satisfactory, however, are those incorporated in the pump-delivery valve itself. The best known of these is the Atlas valve, which is provided with a cylindrical extension placed beneath the valve head and made a plunger fit in the valve guide, so that as the valve returns to its seat an increase in volume takes place on the pipe side of the valve, and thereby reduces the pressure. The action of this valve, however, appears to

be rather more complicated than the above description would indicate; the author has found this valve capable of unloading a system in which the amount of compression of the fuel amounted to 40 per cent more than the volume displaced by the plunger part of the valve. A number of other valves have been produced which enable the system to be unloaded, but it is not the purpose of this work to go into what are really constructional details.

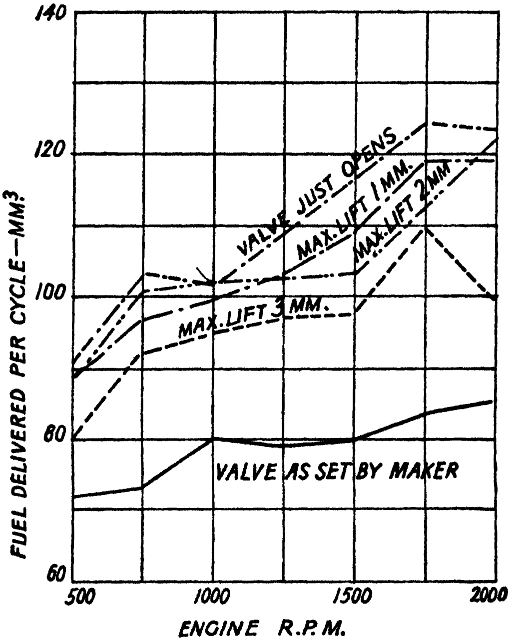
In the author's opinion, the importance of properly unloading the system cannot be over-estimated; the successful operation of the high-speed engine is really dependent upon the system being unloaded down to a very low figure, and if this is not done trouble with the nozzles is certain to be experienced. This does not mean, of course, that nozzle trouble will not be experienced even if unloading is successfully achieved, because nozzle trouble has origins other than a failure to unload, but a combination which when unloaded gives every satisfaction will prove troublesome if pressure is retained in the system. When ascertaining whether a system is unloaded or not, it is not sufficient to examine the conditions at one single speed; the conditions must be examined over the whole range of speeds at which the engine is intended to operate, as it not infrequently happens that over a short portion of the range a considerable pressure is retained, although the unloading may be complete over the rest of the speed range.

15. Speed-delivery Characteristics of the Pump.

The variation of the delivery from the fuel pump with engine speed is a matter of considerable importance with a variable-speed engine. For engines operating at a fixed speed or for marine and aircraft engines the speed-delivery characteristics of the injection system are immaterial, but for engines destined for road transport work, or where full torque is required over a range of speeds, the constancy of the delivery from the pump over the speed range is a matter of first importance. At first sight the jerk pump system would appear to be free from any irregularity in delivery, but this is far from being the case, and wide variations in delivery are frequently experienced at different speeds. It does not necessarily follow that the same delivery at all speeds is best suited to the engine's requirements, and actually the metering characteristics of the pump should match as nearly as is possible the requirements of the engine. This means that at full delivery the pump should deliver at each speed the maximum quantity of fuel that can be burned with a clear exhaust, although if the exhaust does not colour up too rapidly some departure from this ideal may be tolerated in the interests of a more uniform torque curve or simplicity in the pump itself.

Usually, especially when really high speeds are required, it is the delivery at the upper end of the speed range that is the most impor-

tant; owing to the volumetric efficiency of the engine falling off at the higher speeds, the amount of air available for combustion decreases with the speed, and the quantity of fuel that can be burned will therefore be reduced, while at the same time a change in combustion conditions may introduce yet another variable. In the case of engines capable of speeds of above 1500 r.p.m., it is not infrequently found that the quantity of fuel that can be consumed per cycle with a clear exhaust falls off somewhat at the lower end of the speed range, i.e. at



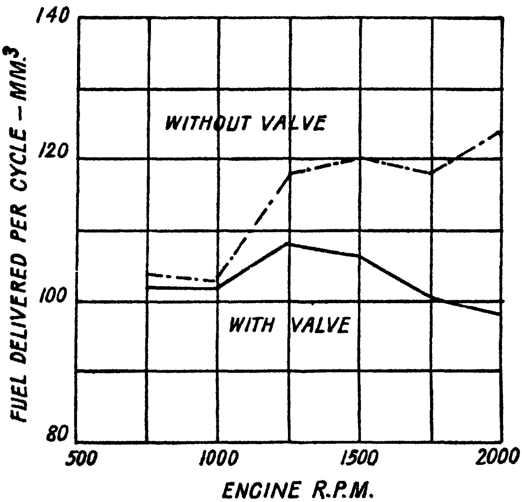
By courtesy of The Automobile Engineer.

Fig. 129.—Influence of an alteration to the delivery valve of the fuel pump

speeds below about 750 r.p.m. Such conditions complicate the situation considerably, and show the necessity for some means of adjusting the speed-delivery curve of the system to meet the requirements of the engine. It may be stated here that such adjustment is possible, but it is not a simple matter, and at present seems to be entirely a matter of trial and error and of luck, although the increasing knowledge of the true nature of the injection process should result in improvements being made in this direction.

The delivery characteristics of the system do not depend upon the pump alone, although this is the most influential factor; the type of nozzle and the length of the pipe both exert their influence. The

delivery from the pump is very greatly influenced by its delivery valve, and a change in the delivery valve may produce a great change in the speed-delivery curve of the pump. This is very well illustrated by fig. 129, which shows the results from a series of experiments with an Atlas valve. The spring was removed and an adjustable stop was fitted so as to limit the lift of the valve, the lift being measured by the distance that the plunger portion of the valve stem rose out of its guide. The delivery normally obtained with the valve as sent out by the maker, measured with the same combination of fuel pipe and injector and delivering fuel to the same engine, is shown also. The



By courtesy of The Automobile Engineer.

Fig. 130.—Effect on delivery of removing the delivery valve from the fuel pump

control rack of the pump was brought to the same position in each case, so that the differences shown are due solely to the alteration made to the delivery valve. The curves indicate that for the conditions of this experiment the delivery of the pump decreases as the lift of the delivery valve increases. It would indicate also that the spring is a very potent factor in controlling the delivery.

The effect of removing the valve entirely is shown in fig. 130. In this instance the delivery from the pump was re-adjusted to the " smoke limit " of the engine at 1000 r.p.m., and it will be seen that the delivery increased rapidly with the speed when no valve was fitted, but with the valve the delivery, after rising slightly up to 1250 r.p.m., thereafter fell steadily as the speed was increased still further.

A point of interest is that whereas when the valve was in position it required a movement of the control rack of 15·5 mm. from the

" stop " position to give the required delivery at 1000 r.p.m., a movement of only 11 mm. was required when the valve was removed. This illustrates the tremendous influence of the valve upon the delivery characteristics.

The nozzle exerts a marked influence upon the speed-delivery curve of the system, this being especially noticeable with fixed-orifice nozzles, which offer an increasing resistance as the speed increases, so that the fuel pressure rises rapidly with the speed. The spill port opens at the same point in the pump stroke and the nozzle valve closes at the same pressure at all speeds, so that the prolonged delivery indicated in figs. 125 and 126 will be curtailed and delivery restricted. Hence, in investigating the speed-delivery curve, it is not possible to consider the pump alone, but the whole system must be considered as a unit. The engine must be considered also, because variations in gas pressure at different speeds will have a bearing upon the ultimate result, and after all it is the quantity of fuel delivered into the engine that is the prime consideration.

16. The Influence of the Fuel Pipe Length.

The relationship between the length of the fuel pipe and the injection lag has already been mentioned when discussing the principles governing the fuel-injection system, but the practical importance of the length of the fuel pipe is very well brought out by figs. 131 to 136, which show the results obtained from some experiments with fuel pipes of different length with three different injector nozzles. In carrying out these experiments the fuel pump timing remained unaltered, the spill port closing in each case at a point 21° before top dead centre. The position of the control rack of the pump was maintained constant also, although the quantity of fuel delivered was not recorded. In each case the fuel was delivered to atmosphere and not into the engine, and the results obtained represent the effects produced by a change in pipe length alone. Under practical conditions a change in pipe length can influence the injection timing in two ways, firstly by the change in the time taken by the pressure wave to travel to and fro along the pipe and build up a pressure sufficient to open the nozzle valve, and secondly by causing a difference in the amount of unloading which takes place. Obviously a change in pipe length which results in a fairly high pressure remaining in the pipe at the end of the injection will result in a reduction in the time necessary to produce the pressure required to open the nozzle valve and vice versa. A progressive change in the length of the fuel pipe does not therefore necessarily mean a change in injection lag which will correspond directly with the change in pipe length. This point is brought out rather well in fig. 131, which indicates that with pipe lengths of less than about 400 mm. the injection timing became somewhat erratic. The unloading of the system is,

of course, intimately connected with the pressure produced during injection, the higher the pressure the greater the possibility of some pressure being retained at the end of injection. The use of an extremely short pipe, by reducing the volume of fuel under compression, tends to increase the maximum pressure and explains why the shorter pipe resulted in the retention of pressure in the system. The nozzle used for the experiment illustrated in fig. 131 was a fixed orifice nozzle, and with the shorter pipe the movement of the fuel would conform more nearly to the plunger displacement, necessitating the production

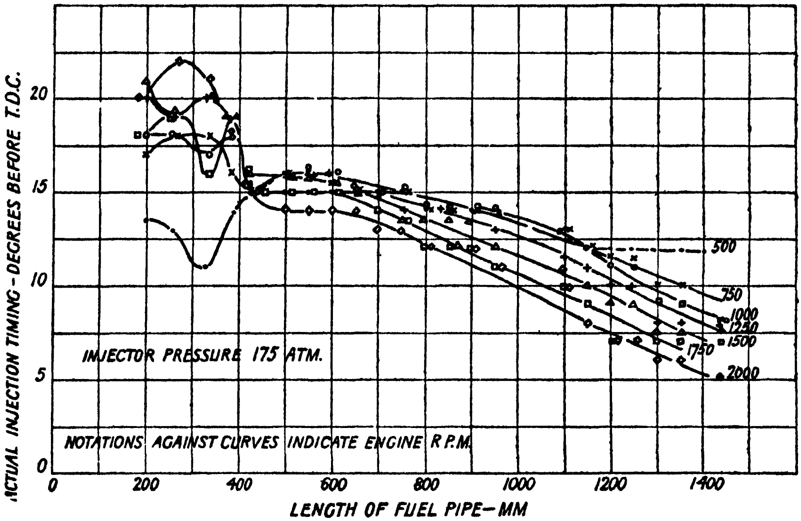


Fig. 131.—Influence of fuel pipe length upon injection timing at different engine speeds: multi-hole nozzle

of higher pressures to discharge the fuel through the orifice. With pipe lengths between about 400 mm. and 650 mm. no very great variation in injection timing took place.

As the length was increased beyond 650 mm. the injection lag became more and more pronounced and increased considerably as the speed increased. It will be obvious that with this particular combination of nozzle, fuel pump and pipe size (the pipe had a bore of 1·5 mm.) the length of the pipe should be kept within the limits of 400 to 650 mm. if the injection timing is to be retained reasonably constant over a good range of speeds. The influence of a change in speed when the pipe length remains constant is shown in fig. 132, which gives some of the same results plotted against a base of speed and again brings out the small amount of variation in the timing with pipes between 400 and 600 mm. long.

That a length of between 400 and 600 mm. is not a fundamental value is shown by figs. 133 to 136, which were obtained with different types of nozzle set at a lower opening pressure and fuel pipes of 2 mm. internal diameter. The same fuel pump as before was, however, used. In these diagrams the irregularity in timing with short lengths does not appear, and at the same time the length over which only a small amount of timing variation occurs is appreciably shorter, being

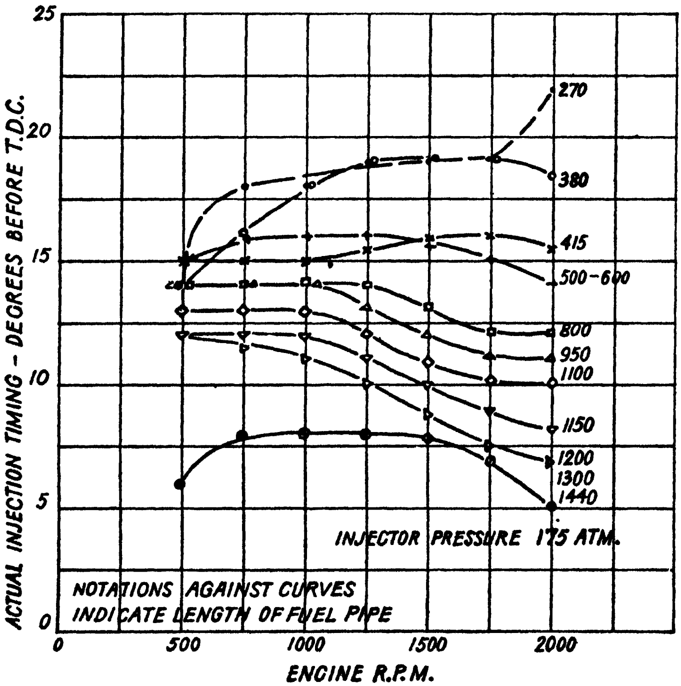


Fig. 132.—Influence of engine speed upon injection timing with different lengths of fuel pipe: multi-hole nozzle

between 300 and 450 mm. with the ordinary pintle nozzle and rather less with the " delay action " pintle nozzle.

From these experiments it will be clear that, although it is not possible to generalize and it is evident that each particular design must be considered on its own merits, the use of fuel pipes of great length, in cases where an engine is intended to operate over a wide range of speeds, will make it impossible to obtain satisfactory operation at all speeds unless some means for adjusting the timing is provided. That this is not realized as fully as it should be is shown by the numerous instances to be seen where pipes of inordinate length are used. When an engine is intended to operate at a fixed speed, the influence of pipe

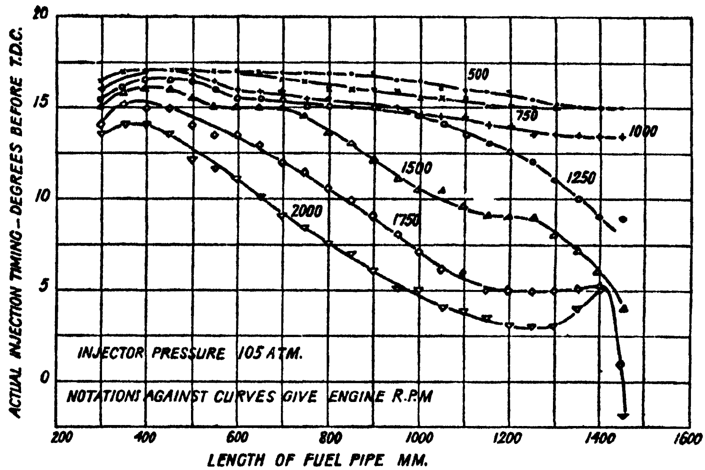


Fig. 133.—Influence of fuel pipe length at different speeds upon injection timing: pintle nozzle

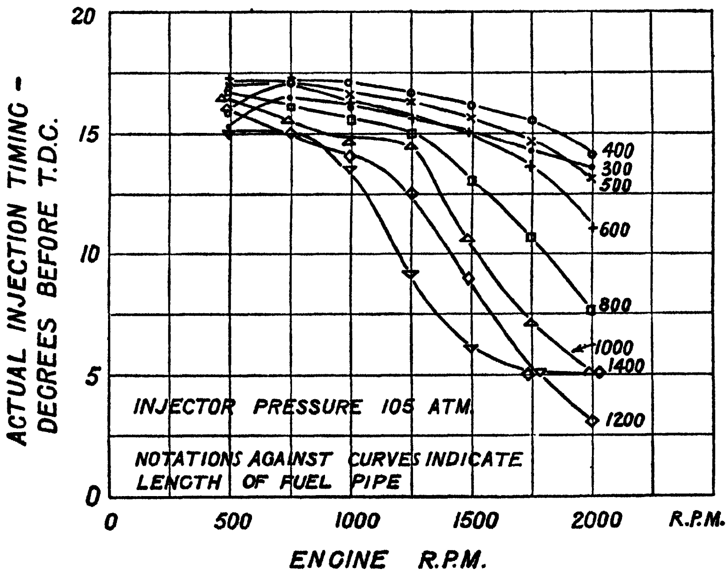


Fig. 134.—Influence of engine speed upon injection timing with different lengths of fuel pipe: pintle nozzle

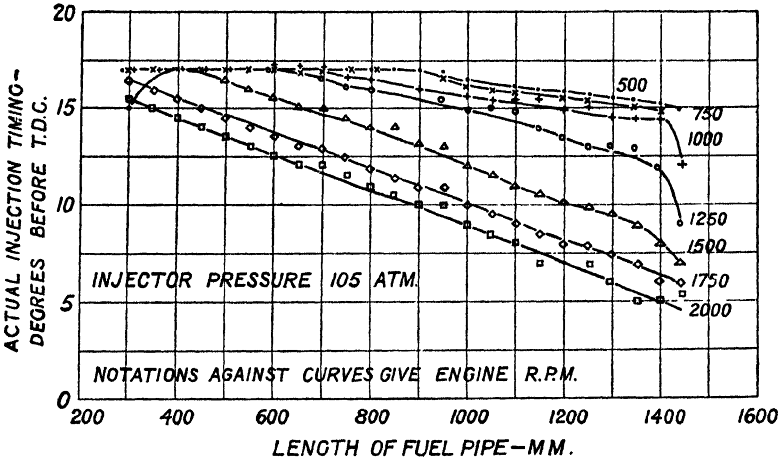


Fig. 135.—Influence of fuel pipe length at different speeds upon injection timing: delay action nozzle

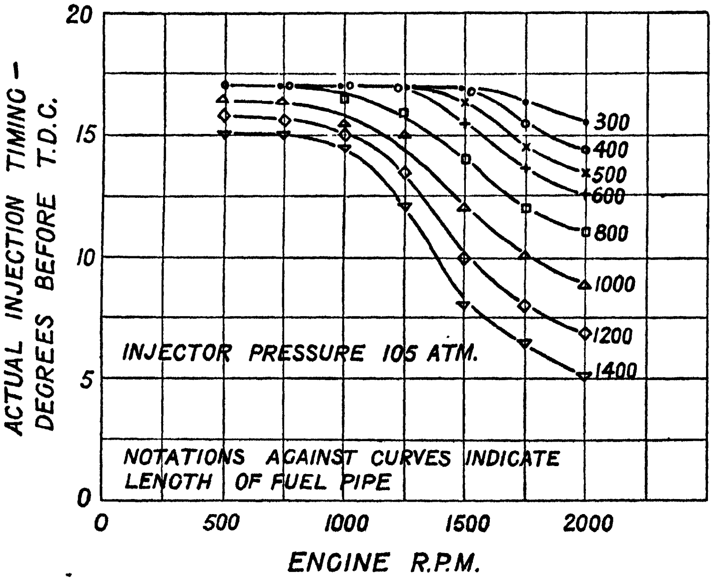


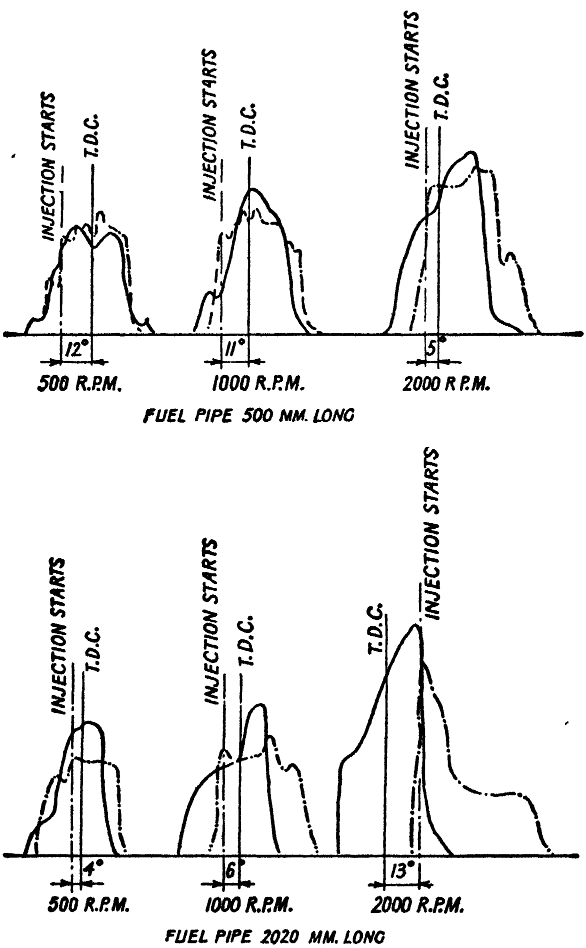
Fig. 136.—Influence of engine speed upon injection timing with different lengths of fuel pipe: delay action nozzle

length upon timing can be allowed for by the setting of the fuel pump, but when a variable engine speed is required it will be impossible to obtain the optimum results at all speeds. To provide an automatic adjustment is an expensive complication, while to leave the adjustment to the operator is most undesirable and leads to unsatisfactory results, especially when the changes in speed are wide and take place frequently. The proper procedure is to lay out the design so that the pipe length will be such as to avoid any undue variation in timing over the desired speed range. In certain instances this may necessitate the division of the fuel pump into two or more units, although, except in the case of large engines, it will seldom be necessary to divide the pump into more than two units. The author is of the opinion that where this enables adequate control over the injection timing to be retained at all speeds the small additional complication is well worth while.

It may be mentioned here that the variation in timing is not the only effect of a long pipe; the actual rate of discharge of the fuel through the nozzle may be altered very considerably by a change in pipe length. The longer the length of fuel pipe, the greater is the difference between the delivery from the pump plunger and the discharge through the nozzle. This point is illustrated by fig. 137, which shows the injection pressure measured at each end of the fuel pipe when pipes 500 mm. and 2020 mm. long were used. This indicates a very material difference in the form of the pressure diagrams from each end of the pipe, and shows that the pipe length influences conditions at the pump end as well as the nozzle end of the pipe. This will be seen from the difference in form of the diagrams even at low speeds. A point of interest is that at 2000 r.p.m. with the long pipe the actual injection takes place almost entirely after the pressure at the pump end of the pipe has fallen to zero. It should be mentioned that the apparatus with which these diagrams were taken would not show properly the more rapid fluctuations in pressure, and although the diagrams for the pressure at the pump end of the pipe represent the conditions fairly accurately, those at the nozzle end of the pipe do not reproduce properly the smaller fluctuations produced by the oscillation of the nozzle valve.

The use of a very short pipe produces a "harder" injection; the increased volume of the longer pipe produces a cushioning effect by an increased compressibility and also by the longer period of time between successive waves. Apart from its influence upon timing, a long pipe is not necessarily detrimental to engine performance, and under certain conditions it may even prove beneficial by modifying the rate of injection, although an improvement obtained by these means is not likely to be maintained over any but a very narrow speed range, so that this is not a suitable expedient to adopt for an engine

called upon to cover a wide speed range. Any modulation produced by a change in pipe length should be capable of production by the adjustment of factors which are independent of dynamical effects,

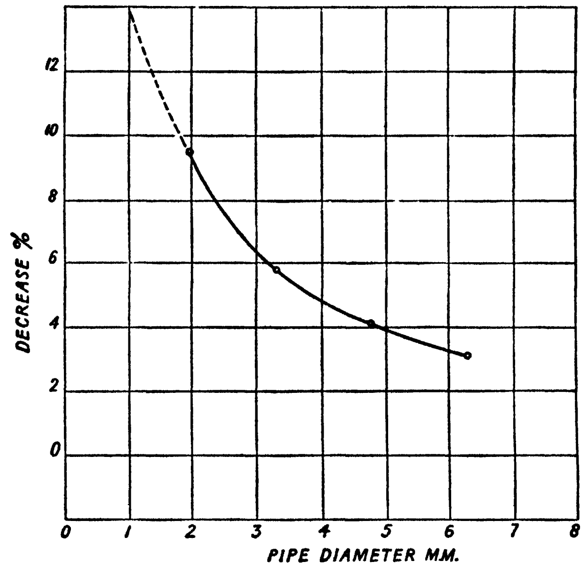


By courtesy of The Automobile Engineer.

**Fig. 137.—Influence of fuel pipe length upon the pressure in the injection system
Full line shows pressure at pump. Chain line shows pressure at nozzle**

and the improvement will then be retained at all speeds. It has to be admitted, however, that the knowledge available at present as to the quantitative effects of modifications made to the injection system is somewhat scanty; the trial and error method has usually to be employed, and this tempts one to include the pipe length in the experiment.

A change in the diameter of the fuel pipe changes the volume of fuel under pressure in the system, and therefore influences the amount of compression of the fuel. The outside diameter of the pipe is usually settled by the connexions provided on the pump and the nozzle holder, so that a change in the internal diameter of the pipe usually means a change in the wall thickness of the pipe and therefore of the expansibility of the pipe. This is equivalent to a change in the bulk modulus of the fuel.



By courtesy of Inst. Mech. Eng.

Fig. 138.—Effect of pipe diameter on speed of propagation. Decrease below value given by $\sqrt{\frac{Kg}{\rho}}$

Apart from the influence of the expansibility of the pipe, the diameter of the pipe has an influence upon the speed of propagation of the sound waves in the fuel, the speed being reduced slightly as the pipe diameter gets smaller. This has been investigated by Talbot,* using pipes of different sizes and fuel at different pressures. Talbot's figures were confirmed by Giffin and Rowe,† who give the curve shown in fig. 138 for the percentage reduction in the speed of propagation below the value given by the equation $V_s = \sqrt{(Kg/\rho)}$ for a pipe pressure of 2000 lb./sq. in. This curve shows that the rate of decrease is rather rapid as the pipe diameter is reduced below 3 mm. For small

* *Phil. Mag.*, 7th series, Vol. 19, p. 1126.

† Pressure Calculations for Oil Engine Injection Systems, *Inst. Mech. Eng.*, Vol. 141, 1939.

high-speed engines the more usual pipe diameters are 1·5 mm. to 2 mm., pipes as large as 3 mm. not being commonly used. With the higher pressures associated with the fixed-orifice nozzle the smaller diameter seems to be preferred, probably because of the smaller amount of fuel subjected to high pressures. With pipes of these diameters a decrease in the velocity of 10 to 12 per cent will be experienced, and from the trend of the curve a reduction of around 14 per cent is indicated for a pipe 1 mm. in diameter.

The author obtained a material improvement in performance from a certain engine when a fixed-orifice nozzle was used by reducing the diameter from 2 mm. to 1·5 mm., the improvement in smoothness when idling being especially marked; but a further reduction to 1·0 mm. resulted in a loss of performance. Too small a pipe results in a big drop in pressure along the pipe, even when the length is comparatively short.

The influence of the length of the fuel pipe must be considered apart from the effects of the compressibility of the volume of the fuel it contains. The length of the pipe governs the injection delay. The rate at which the pressure in the system builds up is affected by pipe length, because although the increase in pressure produced by any individual reflection of the pressure waves is the same for any length of pipe, the time interval between successive reflections increases with the length of the pipe, and the time required to reach a given pressure is therefore longer with a long pipe than with a short pipe. The time to reach the opening pressure of the nozzle is therefore longer and the injection delay increases with the length of the pipe, but, in addition, the build-up of the pressure after the nozzle has opened is at a lower rate than with a short pipe and the rate of injection is therefore reduced. Speaking in general terms, the shorter the pipe and the smaller the quantity of fuel under pressure, the more closely does the flow of fuel through the nozzle follow the displacement at the pump plunger face.

The length of the pipe is a factor in determining whether or not there is any residual pressure at the end of injection, and the intensity of the pressure retained. Basically, the residual pressure is a question of the amount of compression in the fuel, i.e. of the pressure at the end of injection and the volume of fuel under pressure, but residual pressure is more likely to be present with a long pipe than with a short one. For in addition to the increased volume of fuel under pressure, and therefore the increased quantity of fuel that must be discharged from the system to reduce the pressure to zero, the greater time taken by the negative waves to travel to and fro along the pipe and destroy the pressure gives the delivery valve more opportunity to regain its seat before the pressure is destroyed and so to retain pressure in the system.

From almost every point of view, therefore, it is desirable to make

fuel pipes as short as possible. With pipes of really short length it should be possible to reduce the diameter to a figure less than that required when a long pipe is fitted, because the resistance offered by the pipe decreases with length.

Enlargements in the pipe are undesirable, because they increase the compression of the fuel and help to prolong injection after the spill port has opened. The effect of the increase in compression is particularly marked when the enlargement is at the nozzle end of the system, because the re-expansion must take place by the fuel travelling the whole length of the system as the alternative to being discharged through the nozzle. Local enlargements also have the effect of reducing the intensity of the pressure wave, much as the energy of a wave from the sea driven along a narrow channel is dissipated on reaching an enlargement in the channel. The ideal arrangement is for a simple and sudden change from the diameter of the pump down to the diameter of the pipe and from that of the pipe down to the diameter of the nozzle without any other variations in diameter throughout the length of the system. This, however, is not practical, but it should be possible to approach a good deal closer to the ideal than is done in many instances, and careful attention to this matter should result in improved conditions.

17. The Mechanism of Atomization.

The fuel injection nozzle has two functions to perform: (1) it has to break up the fuel into a condition suitable for rapid and complete combustion, and (2) it has to direct the spray in such a manner that, in conjunction with the movement of the air itself, the maximum possible quantity of oxygen is consumed without leaving any unburned, or partially burned, fuel.

It is difficult to say which of the two functions is the more important, for unless the fuel is properly subdivided it cannot be completely burned in the short time available, while if it is not properly directed into the combustion chamber it cannot get into contact with the quantity of oxygen necessary for its complete combustion. The first condition is certainly the one most easily fulfilled, or perhaps it would be more correct to say the one in which the highest degree of efficiency is attained, because, when judged by the quantity of oxygen that can be used by the nozzle's unaided efforts at distribution, the efficiency of the best form of nozzle is deplorably low in a high-speed engine. The distributing abilities of a nozzle are very low, and without the assistance of a suitable air movement the percentage of oxygen that can be used is quite small.

The process of breaking up the fuel into small particles is commonly referred to as *atomization*, although this term is really a mis-

nomer, because nothing approaching atomization in the proper sense of the word is ever attained. Pulverization is probably a better word to describe the process, and has been used in connexion with air blast injection.

In the past it has generally been assumed that the breaking-up of the fuel stream occurs in the nozzle and is assisted by the sudden release of the fuel from a high pressure as it issues from the nozzle. Recent investigations into the mechanism of atomization carried out under the auspices of the National Advisory Committee for Aeronautics, Washington, D.C., have shown that this is not the case, and that the break-up of the fuel stream does not occur until after the fuel has issued from the nozzle, and is performed by the friction between the air and the rapidly moving jet of fuel. The fuel issues from the nozzle as a solid jet at a high velocity, and the interaction between the fuel and air results in ligaments of fuel being torn from the stream of fuel, this action being so rapid at the velocity at which the fuel stream issues from the nozzle that even with air at atmospheric pressure the complete disintegration of the jet takes place within a short distance of the nozzle. The fuel particles do not remain in the form of ligaments, but are contracted by surface tension into globules, the process being almost instantaneous. Under the conditions which exist in the combustion chamber the action is even more rapid, on account of the much greater density of the air.

Experiments by D. W. Lee and R. C. Spencer,* who by the illumination of an electric spark photographed the fuel sprays produced by the discharge of fuel under different pressures into atmospheres of different densities, showed that the fuel issues from the nozzle as a solid stream, and that the break-up distance varies with the pressure of the fuel and the air, and also that at reduced pressures the jet does not break up until it has travelled a considerable distance from the nozzle. They also showed that the extent of the break-up of the jet increases with increasing pressure, and at high pressures it was not possible to obtain satisfactory photographs because the great mass of particles resulted in the image being blurred.

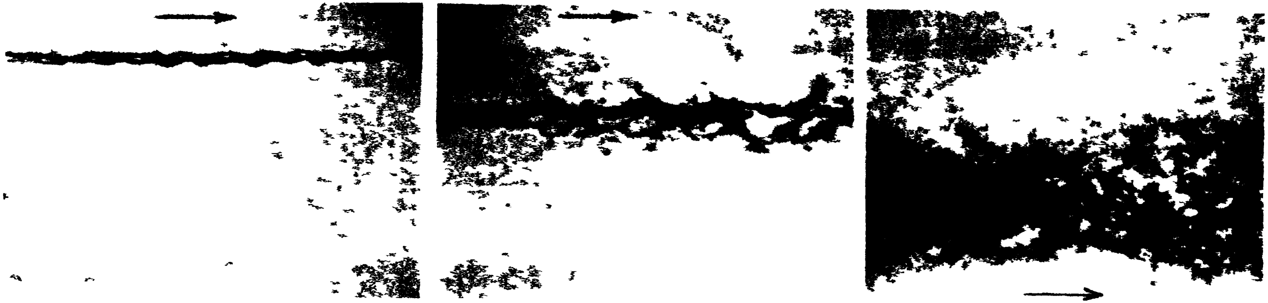
Amongst the conclusions of Lee and Spencer the following are of particular interest:

"At a given distance from the orifice, the disruption of the jet and the dispersion of the fuel increase with an increase in the jet velocity or an increase in the air density."

"At a given value of jet velocity and air density, the disruption of the jet and the dispersion of the fuel increase with the distance from the nozzle until the relative velocity between the fuel and the air becomes so low that the air no longer tears the fuel apart."

* Preliminary Photomicrographic Studies of Fuel Sprays, *Technical Note* No. 424, *Langley Memorial Aeronautical Laboratory.*

PLATE I



Air density 0 00526 atmosphere   Air density 1 0 atmosphere   Air density 13 3 atmosphere

Injection pressure 1000 lb sq in   Distance from nozzle 1 in   Discharge orifice diameter 0 008 in

At the nozzle   in from nozzle   in from nozzle

Injection pressure 1000 lb sq in   Air density 1 atmosphere   Discharge orifice diameter 0 014 in

FIG 139—THE PROCESS OF ATOMIZATION

R. A. Castlemain, Jr.,* in summarizing a discussion on the mechanism of atomization, states "(1) that the atomization accompanying solid injection occurs at the surface of the liquid after it issues as a solid stream from the orifice; and (2) that such atomization has a mechanism physically identical with the atomization which takes place in an air stream, both being due merely to the formation, at the gas-liquid interface, of fine ligaments under the influence of the relative motion of gas and liquid, and to their collapse under the influence of surface tension to form the drops in the spray."

The break-up of the fuel into the minute particles necessary for satisfactory combustion is thus attributed not to any disruptive effect produced by the fuel passing at a high velocity through the nozzle orifice, but to the friction between two fluids, the fuel and the air. The reaction between the two is shown very clearly by photographs published by Lee and Spencer (*loc. cit.*), representative samples of which are shown in figs. 139 and 140, and on Plates I and II, which are reproduced from the *Technical Note* referred to above. The photographs in fig. 139 (Plate I) show very clearly the effects of a change in air density upon the spray produced from a certain nozzle at a fixed distance from the nozzle, while fig. 140 (Plate II) gives an excellent idea of the process of disintegration of the jet, showing the stages passed through before the actual globules of fuel are produced.

From these experiments it appears that the structure of the jet of fuel changes with the distance from the nozzle. Close to the nozzle the fuel issues as a solid stream, but at a short distance from the nozzle the action of the air tears ligaments of fuel from the solid stream, these ligaments immediately resolving into globules. The solid jet of fuel is thus surrounded by a mass of small particles, a layer of ligaments immediately surrounding the solid core, while a layer of globules surrounds the ligaments. As the distance from the nozzle increases the solid core decreases and finally disappears entirely, leaving nothing but a cloud of particles in globular form. Confirmation of this is obtained in the numerous photographs published by the National Advisory Committee for Aeronautics of the sprays in the engine combustion chamber under running conditions. In hardly any instance is the course of the main body of the spray shown to be deflected from the straight line; only the envelope of the spray is shown deflected. In *N.A.C.A. Report* No. 520 it is stated that "tests showed that air moving at 60 ft./sec. or less will blow the envelopes away from the cores of sprays during the injection period". Further confirmation is to be found in the fact that the marks of fuel striking the side of the combustion chamber remote from the fuel nozzle are always directly opposite to the nozzle orifice, even when a high swirl velocity is used as in a Ricardo Comet engine. There are indications of the lighter

* *Report* No. 440, *National Advisory Committee for Aeronautics*, Washington, D.C.

stuff being carried off downstream, but the marks indicative of fuel of any mass striking the walls are always directly in line with the nozzle orifice. If the fuel were broken up into globules inside the orifice this could hardly be the case, unless the spray contained a considerable body of really heavy particles too heavy to be deflected by the air stream, a most unlikely event in a spray sufficiently finely divided to be burned cleanly and quickly enough to give a good performance in a high-speed engine. As will be seen later, experiments to determine the size of the fuel particles in the spray show that the great majority of the particles are of exceedingly small dimensions.

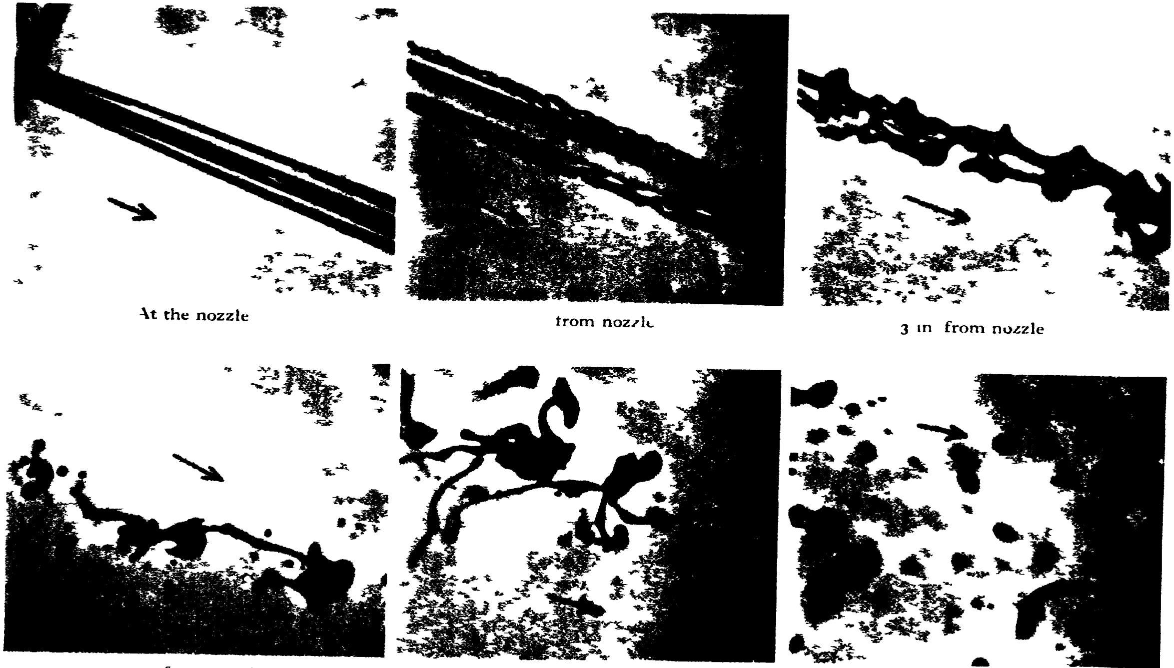
The size of the fuel particles is a matter of importance, because not only is the time required to ignite the globule influenced by its size, but so too is the time necessary for its complete combustion. The mass of the individual globule influences the distance it will travel, but the importance of this point is reduced by conditions introduced by the large number of globules. The mass of, and therefore the kinetic energy contained by, any individual globule is exceedingly small, while the resistance offered by the air is high, especially with the increased density of combustion chamber conditions. The energy of any individual globule is therefore soon absorbed and its velocity destroyed, and taken alone the globule would not travel far from the spot at which it became detached from the stream of fuel. Kircher * shows that under combustion chamber conditions a single droplet of fuel could in no case travel more than 1 in. The energy lost by the globules is, however, transferred to the air which opposed their movement, with the result that a motion is imparted to the air, the latter motion being in the same direction as that in which the globules themselves were travelling, so that some air and the fuel globules travel along together, the mass effect of the fuel and air enabling the fuel to travel a distance that could never have been attained by an individual globule.

The resistance of the air, together with the eddying of the air, cause a bushing-out of the spray, and this bushing-out is increased considerably as the density of the air increases, and at the same time an increase in density will decrease the maximum distance travelled by the spray.

The break-up of the fuel stream into a spray being the result of friction between the jet of fuel and the air, it is reasonable to assume that as the velocity of the jet is increased so the rate at which the jet is broken up will increase, and that the fineness of the spray will be improved also. The velocity of the jet is governed by the pressure drop across the orifice, so that the fineness of the spray should increase with an increase in injection pressure. Similarly, for a given pressure drop across the nozzle a reduction in diameter of the orifice will increase the surface-volume ratio exposed to the action of the air, and

* The Atomization of Liquid Fuels, *T. M.*, No. 331, *N. A. C. A.*

PLATE II



At the nozzle

from nozzle

3 in from nozzle

5 in from nozzl

, in from nozzle

10 in from nozzle

Injection pressure 1000 lb sq in Air densit 1 atmosphere Di charge rifice diameter 0 020 in

FIG 140 —THE PROCESS OF ATOMIZATION

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improved atomization is to be expected. Under practical conditions, other things being unaltered, a reduction in orifice diameter results in an increase in pressure drop across the nozzle, so that there is a two-fold change in conditions.

If the atomization of the fuel takes place after the fuel has left the orifice, it is reasonable to suppose that the length of the orifice has no direct influence upon atomization. As the length of the orifice may influence the coefficient of discharge, however, the velocity will be affected, with a corresponding influence upon the atomization. The same will, of course, apply to any modification of the entry or exit edges of the orifice which produces a change in the discharge coefficient.

Researches into the fineness of the fuel spray have been conducted by a number of investigators. Two methods have been employed: (1) measuring the time required for the drops to fall a certain distance, and (2) collecting the drops on a suitable surface and examining the surface under a microscope. The former method has the advantage of simplicity, but does not lend itself readily to determinations under conditions of increased air density. With drops of such small dimensions as those of a fuel spray the rate of fall in still air is constant, the velocity of fall depending upon the diameter of the drops. The time of fall being known, the drop diameter can be calculated from Stokes' law as follows:

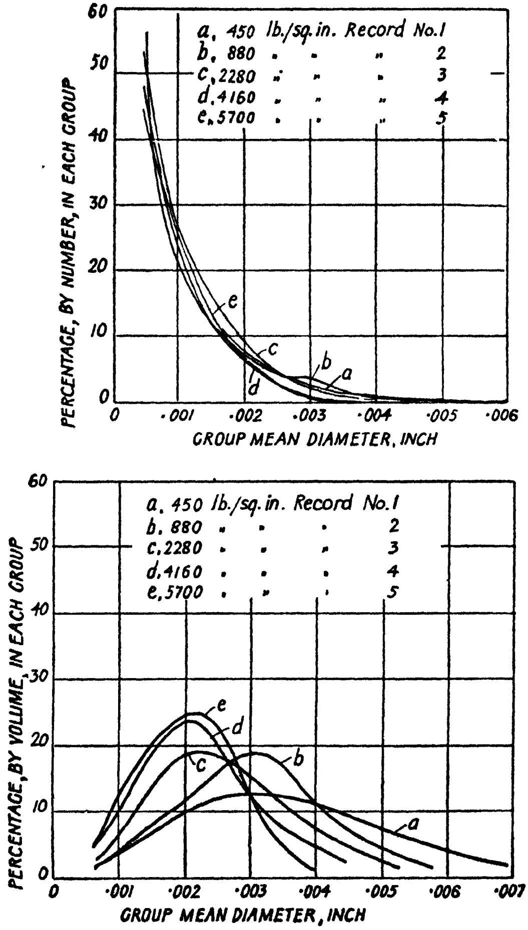
$$V = \frac{2}{9}\left(\frac{\rho_o - \rho_a}{\mu}\right) r^2 g,$$

where V is the velocity of fall, ρ_o the density of the oil, ρ_a the density of the air, μ the viscosity of the air, and g the acceleration due to gravity.

With the second method it is possible to make measurements under pressures representative of actual engine conditions. It is rather more tedious, but at the same time is probably more accurate. Probably the most comprehensive series of experiments in this direction are those described by D. W. Lee.* The spray particles were collected on a carefully smoked glass screen; certain zones were examined under the microscope and the size of the globules measured and the numbers counted. The effects of injection pressure, chamber density, orifice diameter, and length-diameter ratio of the orifice were investigated, as well as different types of nozzle. In measuring the sizes of the droplets steps of ·0005 in. were taken, and the drops grouped to the nearest ·0005 in. in size, and in analysing the results curves were plotted in which both the percentage of the total number of drops falling into any one group and the percentage by volume falling in any one group were plotted against the group mean diameter.

* *Report* No. 425, *National Advisory Committee for Aeronautics.*

These experiments showed that with pressures above 2280 lb./sq. in. and a single-hole orifice of diameter 0·020 in., the largest percentage of fuel by volume consisted of particles having a group mean diameter



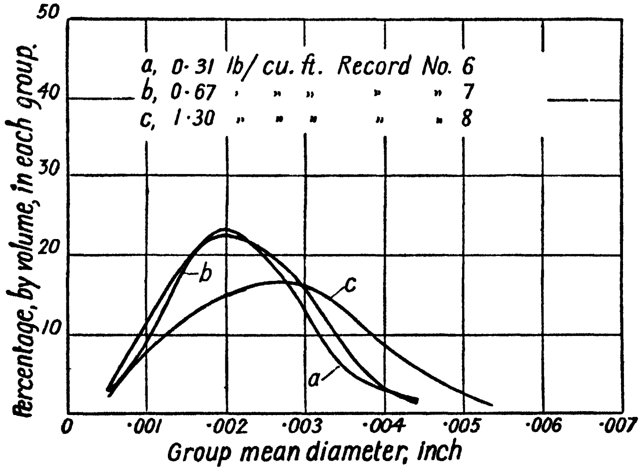
By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

Fig. 141.—Atomization curves for sprays from a nozzle having 0·020-in. orifice, injected at different pressures

of 0·002 in., the actual quantity rising from just under 20 per cent at a pressure of 2280 lb./sq. in. to 25 per cent at pressures of 4160 and 5700 lb./sq. in. With the higher pressures the group mean diameter did not exceed ·004 in., the distribution each side of 0·002 being remarkably uniform. With a pressure of only 880 lb./sq. in. the group

mean diameter representing the maximum volume of fuel was 0·003 in., and represented just under 20 per cent of the total, while the maximum group mean diameter increased to 0·006 in., the distribution each side of 0·003 in. again being remarkably uniform.

The results of this series of experiments are shown in fig. 141, reproduced from Lee's report, and indicate that as the injection pressure is increased from a low value the spray tends to become finer and more uniform until a certain pressure is reached, beyond which conditions become stabilized. At pressures in excess of a little over 2000 lb./sq. in., the mean group mean diameter on a volumetric basis



By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

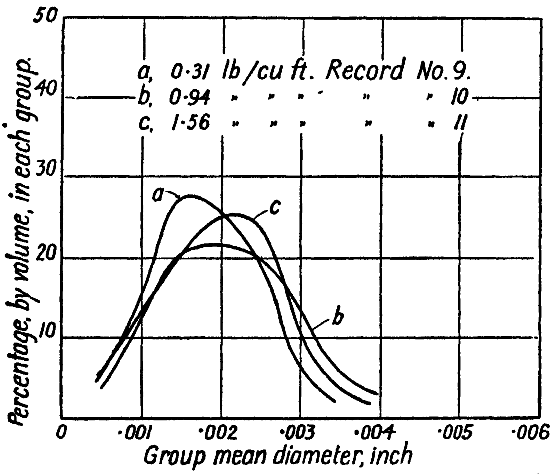
Fig. 142.—Effect of chamber-air density on fuel atomization. Effective injection pressure, 4100 lb. per sq. in.

appears to have a constant value of about 0·002 in. Numerically, of course, the number of globules in each group increases rapidly as the group mean diameter decreases, as is shown by fig. 141, which gives not only the volumetric but also the numerical distribution. The volumetric distribution curve gives a much clearer idea of the influence of the pressure.

The influence of a change in air density is shown in figs. 142 and 143. The results are not so consistent as those obtained for a change in pressure, and the inference is that the effect of density is not so marked as is usually assumed. It is possible that density affects the distance from the nozzle at which atomization takes place rather than a change in the fineness of the atomization. The chamber used in the N.A.C.A. experiments was 18 in. long and allowed plenty of distance for spray penetration. This contention is supported by the evidence of fig. 144, where it will be seen that the higher the density of the air the closer

to the nozzle is the bulk of the fuel deposited on the screen, although, of course, the stopping power of the air increases with the density. It is interesting to note, however, that even with air at a density of 1·30 lb./c. ft. quite a large percentage of the fuel reached the end of a chamber 18 in. long.

The effect of orifice diameter is seen in fig. 145 (p. 282), which shows that the atomization becomes finer and more uniform as the orifice diameter decreases. In these tests the pressure across the nozzle was the same in each case. The marked increase in fineness for an orifice of diameter 0·008 in. as against 0·020 in. is very striking, and agrees



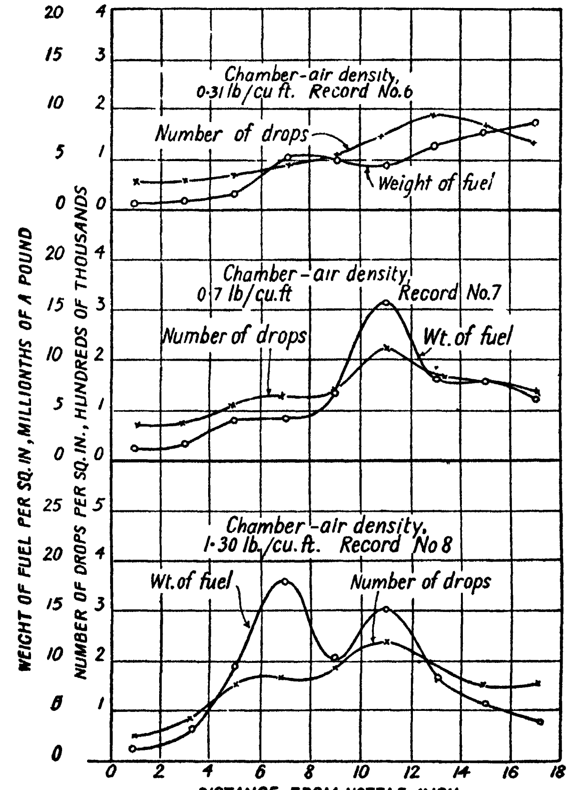
By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

Fig. 143.—Effect of chamber-air density on fuel atomization. Effective injection pressure, 4140 lb. per sq. in.

with the author's experience that an engine with an open combustion chamber is much more sensitive to a change in orifice diameter than to any other change to the nozzle, an increase from a diameter of 0·010 in. to 0·012 in. in one case being sufficient to spoil the engine performance completely, and in another case a very small discrepancy in the orifice diameter producing a big loss of performance. Under engine conditions, a change in orifice diameter results in a change in the pressure drop across the orifice and introduces an additional factor, whereas in the N.A.C.A. experiments the pressure was maintained constant.

The effect of a change in length-diameter ratio is seen in fig. 146 (p. 282), which shows that a change in length-diameter ratio of from 0·5 to 6·0 has practically no effect upon the atomization of the fuel. As the ratio increased, however, there was a slight increase in penetration, as will be seen from fig. 147. This again is borne out by the author's experience, when an increase of l/d from 1·6 to 5·0 produced no change

in performance in the case of an open combustion chamber, using a four-hole sprayer. From this it may be inferred that the changes in engine performance which are noted in certain instances with a change in length-diameter ratio are due to change in penetration and also to the change in pressure produced by the altered resistance resulting from the change in length.

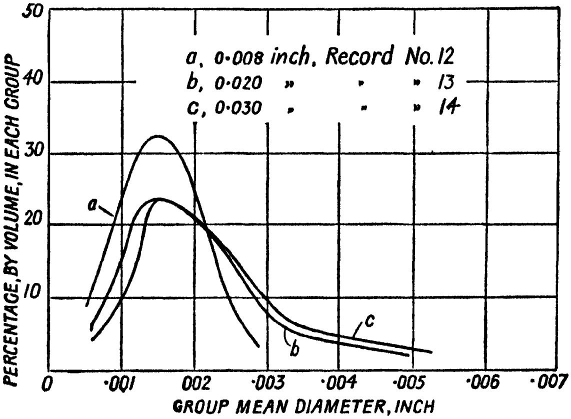


By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

Fig. 144.—Effect of chamber-air density on spray penetration. Effective injection pressure, 4100 lb. per sq. in.

From these experiments it would appear that the change in penetration is not very marked, which suggests that the big changes in engine performance which are sometimes produced with an alteration in orifice length may be due to a change in pressure, and therefore in jet velocity. At the same time the fact that a small change in penetration may, under certain circumstances, produce a large alteration in performance should not be overlooked. By removing fuel from an overcharged zone in the combustion chamber to an undercharged one,

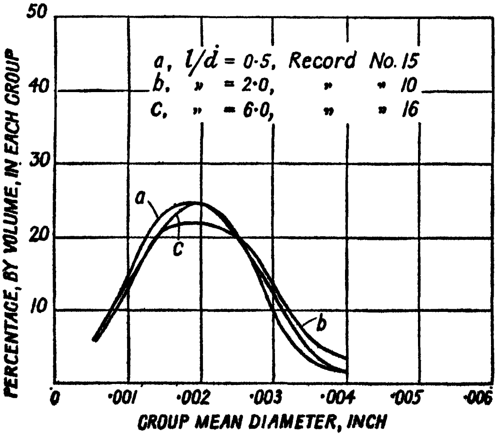
quite a small modification in distribution may produce quite a large improvement in engine performance and vice versa. The orderly air



By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

Fig. 145.—Effect of orifice diameter on fuel atomization. Mean effective injection pressure, 3913 lb. per sq. in.

movement necessary for a compression-ignition engine and the early completion of combustion offer very little opportunity during combustion for the readjustment of faulty fuel distribution.



By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

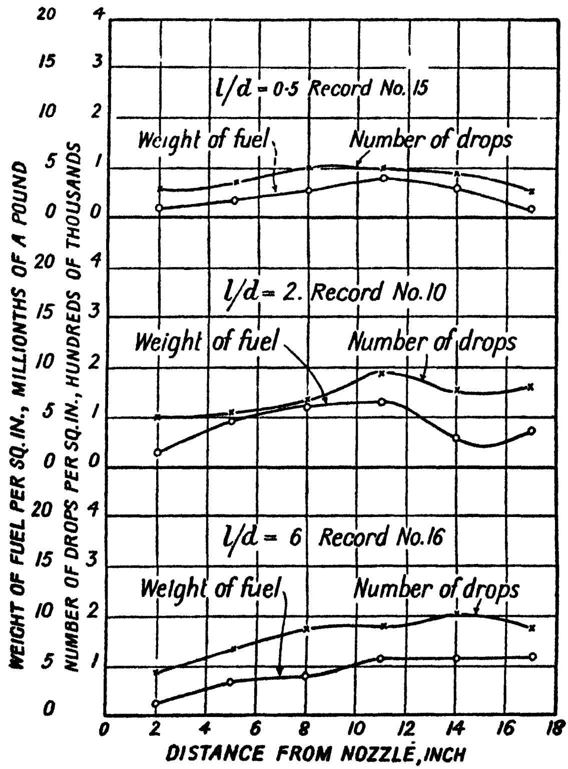
Fig. 146.—Effect of orifice length-diameter ratio on fuel atomization. Mean effective injection pressure, 4133 lb. per sq. in.

Amongst the conclusions drawn from these experiments the following may be quoted from the *Report*:

"Each spray is composed of several million fuel drops whose

diameters vary from less than 0·0025 in. up to 0·0050 in. and sometimes more. By far the greatest number of drops have diameters of 0·0010 in. or less, but those between 0·0015 in. and 0·0025 in. usually contain more than half the weight of the fuel charge.

" When the velocity of the fuel through the nozzle is increased, either by raising the pressure or by improving the design of the injec-



By courtesy of National Advisory Committee for Aeronautics, Washington, D.C.

Fig. 147.—Spray penetration curve for nozzle with different orifice length-diameter ratios. Mean effective injection pressure, 4133 lb. per sq. in.; chamber-air density, 0·94 lb. per cu. ft.

tion system, there is a reduction in the relative number of large drops. The result is a more uniform atomization and a smaller mean drop size.

" A decrease in the orifice diameter also results in a more uniform atomization and a smaller mean drop size.

" The density of the air into which the fuel is injected has little effect on the final atomization attained.

" Within the range of orifice sizes and operating conditions commonly used, the variation in the mean drop size is small. The factor

having the greatest effect on the atomization is the velocity of the fuel as it leaves the orifice, the increase in velocity resulting from an increase in the injection pressure from 2280 to 5700 lb. per square inch causing a reduction of only 20 per cent in the volumetric mean drop diameter."

In view of the statement that each spray is composed of several million fuel drops, the following example will be of interest. The maximum fuel charge of a certain high-speed engine amounts to ·0055 c. in. If we take the average size of the droplet to be 0·002 in. diameter and n as the number of droplets, then we have

$$\frac{4}{3}\pi(\cdot 001)^3 n = \cdot 0055,$$

whence $n = 1{,}311{,}000$; but 0·002 is the average size and the number of quite small particles will make the actual number present several times this figure, as will be seen by reference to fig. 141 (p. 278).

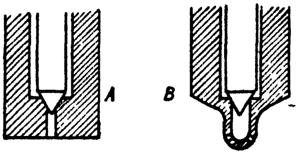


Fig. 148.—To show how the type of nozzle influences the nature of the orifice. A, chamfered-edge orifice. B, square-edge orifice.

The coefficient of discharge of the orifice is a matter of importance because of its effect upon the velocity of the jet as it leaves the nozzle, or conversely upon the pressure required to discharge the fuel at a given rate through the orifice. As with all orifices, the coefficient of discharge of the nozzle is greatly influenced by the shape of the orifice on its upstream side, a rounding or chamfering of the edge of the orifice giving a much higher value than a square edge.

In the great majority of cases the dimensions and construction of the nozzles used for high-speed engines are such that it is not possible to make the nozzles other than with a square or substantially square edge. An exception to this exists in the case of an orifice having a single hole placed axially in the nozzle, this arrangement allowing some modification to the upstream edge of the orifice. With these nozzles a square edge is the exception rather than the rule, because the nozzle valve is usually made to seat at the inner end of the hole, and the form of the seat, being conical, automatically eliminates the square edge as shown at A (fig. 148). With a multi-hole nozzle such as shown at B or when a single hole is drilled at an angle with the axis of the nozzle, the physical dimensions of the nozzle preclude anything but a square edge.

The coefficient of discharge for a square-edged orifice of the sizes associated with compression-ignition engine nozzles appears to be in the neighbourhood of 0·7, but there is evidence to show that the figure changes with the pressure on the downstream side of the orifice, in-

creasing with the pressure of the air into which the fuel is delivered.* This condition, however, was for a fixed pressure on the fuel and an increasing air pressure, but under engine conditions the pressure on the fuel increases as the back pressure increases, and so tends to keep down the ratio of the two pressures. The presence of the nozzle valve has no appreciable effect upon the value of the coefficient of discharge so long as the lift of the valve has reached a point at which the restriction offered by the valve is small compared with that of the orifice. In practice it is found that the lift of the valve to provide an area equal to that of the orifice is very small indeed, so that at a very low lift the resistance of the nozzle valve becomes a small part of the total resistance of the nozzle. To quote an instance: the nozzle of a certain engine has four holes of diameter 0·25 mm., while the diameter of the inner end of the valve seat is 2 mm. The lift of the valve to equal the area of the holes will therefore be

$$\pi \times 2 \times l = \frac{\pi}{4}(0{\cdot}25)^2 \times 4,$$

whence $$l = \frac{1}{32} \text{ mm.} = 0{\cdot}00125 \text{ in. (approx.)};$$

the valve had therefore to be only just off its seat to give an opening equal to that through the orifice. A lift sufficient to produce an area appreciably greater than that of the orifice will, of course, be necessary before the influence of the valve disappears, but the lift of the valve necessary to give this condition will be very small indeed.

18. Spray Characteristics and the Combustion Period.

From the foregoing it will be clear that, in general, sprays are much the same regardless of the nozzle or the pressure conditions by which they are produced. Basically, they all consist of fuel particles of about the same size, the chief difference produced by any change in one of the governing factors being in the proportions of the total fuel charge represented by particles of a given size. A higher pressure results in a more uniform spray by reducing the number of larger particles, and also, to some extent, by reducing the maximum particle size and increasing the number of smaller particles. The fact that even quite a low injection pressure produces a large number of the smallest size of particle means that a variation in injection pressure will not have any marked influence upon the delay period.

To quote an instance, under certain conditions the delay period was found to be 10½° with the nozzle valve set to open at 100 atmospheres, while at the same speed and load the delay period was reduced by only half a degree to 10° when the injection pressure was increased

* *N. A. C. A. Report*, No. 373.

to 250 atmospheres. The fact that the delay period remains practically unchanged by a change in injection pressure does not necessarily mean that no change is produced in engine operating conditions. Quite commonly it is found that any appreciable increase in injection pressure is accompanied by a noticeable increase in roughness. Rough running is not the product of the delay period alone; primarily it is due to too rapid a rate of pressure rise, and this can be produced either by a long delay period or too rapid a rate of burning after quite a short delay period. The rate of burning during Ricardo's Stage 2 is very greatly influenced by the fineness of the fuel particles, and for a given delay period any alteration which results either in a finer spray or in a larger proportion of the finer particles will produce a more rapid rate of burning and, by producing a greater rate of pressure rise, cause a loss in smoothness. The finest particles of the spray will govern the delay period, because they will most quickly reach the ignition point. In addition, it is the finest particles that are most likely to be vaporized, so that the amount of vaporization, if any, will be some function of the quantity of very fine particles, and hence the rate of burning tends to be greater as the fineness of the spray is increased. The rough running consequent upon an increase in nozzle opening pressure is therefore due not to a change in the delay period but to a change in spray characteristics which results in more rapid burning.

It must not be assumed, however, that a reduction in delay cannot be produced by an improvement in atomization. A very coarse spray, such as is produced by an open nozzle at low engine speeds, and at normal engine speeds also at the beginning of the injection period before the pressure has built up to a figure sufficient to produce proper atomization, will give a long delay period, because under conditions of low pressure the fuel is broken up first into large particles, the very fine spray not being produced until later. When ignition takes place, the coarseness of the spray results in a relatively slow rate of burning and smoother running is obtained. Under these conditions an improvement in atomization will result in a reduced delay but may at the same time cause some loss in smoothness. Given satisfactory atomization at the start of injection, however, a change in injection conditions which results in a finer spray may be said to have no influence upon the delay period.

A finer spray may, under certain conditions, result in a noticeable improvement in engine performance. The increased rate of burning will increase the constant volume burning, and also will result in combustion being completed earlier in the cycle and thus improve the efficiency. Another point, and one which is of special importance in the vehicle engine, is that the use of a fine spray helps to reduce the odour of the exhaust gases. A coarse spray has a greater tendency to produce products of partial combustion and give a bluish tinge to the

exhaust gases. The elimination of these products results in a marked reduction of the smell from the exhaust.

In order to avoid the formation of coarse spray the pressure at the beginning of injection should be such that adequate spraying is obtained immediately. This necessitates the sudden application to the orifice of the necessary pressure, a condition which is not feasible with the open nozzle, but it is equally important that there should be a sudden cessation of pressure at the end of injection, because a gradual fading away of the pressure will lead to the formation of large particles, and these will be as active, if not more so, in producing an evil-smelling exhaust as those produced at the beginning of the injection.

19. Types of Sprayer Nozzle.

The sprayer nozzle may be classified in several different ways, e.g. according to whether (1) the orifice is fixed or variable in area; (2) the orifice is an open one or is closed by a spring-loaded valve; (3) the spring-loaded valve, when fitted, opens outwards or is of the differential inward-opening type; (4) the nozzle has a single hole or is of the multi-hole variety.

All nozzles will, of course, come under more than one of these categories, and the possession of a certain feature will, in some instances, result automatically in the inclusion of certain others. An open nozzle, for example, is of necessity a fixed-orifice nozzle, while a variable-orifice nozzle has a single hole and cannot come within the open class.

The class of nozzle used will be governed first of all by the type of combustion chamber, and secondly by the nature of the service required from the engine. Speaking in a general way, a compact combustion chamber provided with a vigorous swirl will require only a single-orifice nozzle, while a less compact chamber or one having a more moderate air movement may require two or more sprays in order to make effective use of the available oxygen. With the comparatively small sizes of cylinder associated with really high rotational speeds it is rare for more than four sprays to be used, and either one spray or four sprays appears to be the most common arrangement, two and three holes being much less frequently used, while as many as six holes have been used on occasion.

Where a long chamber is used and the spray therefore has to travel a considerable distance to reach the furthermost point, a single-hole nozzle giving a narrow cone of dispersal is usually required in order that the necessary penetration of the fuel may be obtained.

The service required from the engine enters into the selection of the nozzle owing to the necessity for the nozzle to be adaptable to the changes of load and speed of the engine. This really affects only the choice between an open or a closed nozzle.

20. Types of Fuel Spray.

The sprays themselves may be classed under two headings: hard sprays and soft sprays. The former, as the name implies, have a relatively large amount of energy in the spray, while the latter exhibit the opposite characteristic. Whether a spray is hard or soft does not basically depend upon whether the spray is composed of large or small particles or is produced by a high or a low pressure. The difference is really in the dispersion of the fuel as it leaves the nozzle. In connexion with the process of atomization it was explained that the fuel issues from the nozzle as a solid stream of fuel, and that this is torn apart by the friction between the air and the fuel stream. A nozzle which allows the fuel to issue as a solid cylinder, or several solid cylinders, of fuel will produce a concentrated and compact spray and at the same time will have a solid core of fuel extending for some distance away from the nozzle. The energy in the spray will therefore be concentrated over a relatively small area around its axis and the spray will be "hard". On the other hand, if the fuel issues in the form of a thin hollow cone, as will be the case with an outward-opening spring-loaded valve, the dispersion of the fuel will increase rapidly with the distance from the nozzle, and except for the vicinity of the nozzle itself, the energy will be distributed over a wide area and the spray will be "soft". The degree of hardness or softness will depend upon the pressure drop across the nozzle, which is not necessarily the same as the difference in pressure between the fuel pipe and the engine chamber. The presence of helical guides for the purpose of giving the fuel stream a rotary movement as it leaves the nozzle, and so assisting in the break-up of the stream, will result in a material loss in the pressure available across the nozzle. Such nozzles, which have never been very successful in service, can produce a soft spray from a cylindrical-hole nozzle, and it must not, therefore, be assumed that a cylindrical hole of necessity denotes a hard spray.

21. The Open Nozzle.

For reasons already given (p. 257), the open nozzle has had only a very limited application, and in view of its extreme simplicity it is to be regretted that circumstances do not favour its wider adoption. Two outstanding cases where the open nozzle has been used with success for many years are the Ricardo "Vortex" engine and the Junkers engine.

In the Ricardo engine the orifice takes the form of a single drilled hole arranged coaxially with the nozzle holder and provided with a simple ball non-return valve for the purpose of excluding gas from the system. This non-return valve is situated some distance away from the orifice, and the passage between the two is filled with fluted

spindle to increase the capillary action and so retain the fuel in this part of the system and prevent gas from coming into actual contact with the non-return valve, on the grounds that at high pressure it is much easier to make a valve absolutely tight against a liquid than against a gas. One great advantage of a nozzle in this form is the much smaller diameter necessary to accommodate it, about half of the minimum diameter required by a closed nozzle being all that is necessary for the Ricardo open nozzle. The spray from this nozzle is hard, having a pronounced solid core of considerable length.

The Junkers nozzle is designed to produce a flat spray having a form somewhat like the flame from the old-fashioned gas burner. In principle, the nozzle consists of two parts, the first being an outer body, or thimble, in the form of a hollow cylinder closed at one end. The inner face of the closed end is made conical in form, and the thickness at the centre is brought down to the length required for the orifice, and the orifice, which has a relatively large diameter, is then drilled axially through the centre. The second part is in the form of a plug which fits closely inside the thimble. The end of the plug is conical and fits closely against the conical end of the thimble. A fuel feed passage is drilled axially along the plug and terminates a short distance from the conical end; across the conical face at the end of the plug two shallow grooves are machined so as to lie diametrically opposite one another and meet at the apex of the cone. These two grooves are placed in communication with the axial supply passage by small holes drilled normally to the face of the cone from a point near its base and running into the central passage. The two parts when assembled together form a structure similar in principle and action to an acetylene gas burner; the fuel passing along the passage drilled centrally along the plug first moves outwards to the base of the cone and into the two grooves. The presence of the plug in the thimble makes the two grooves into closed passages along which the fuel flows in two streams which meet at the apex of the cone and, striking against one another, pass out in the form of a flat fan-shaped spray through the central orifice drilled in the thimble. No non-return valve is used with this nozzle.

A more recent example of the open nozzle is that used for the General Motors two-stroke engine. In this design the pump and nozzle are built into a single unit and the nozzle itself is of the multi-hole type, six holes being used. Immediately inside the nozzle is a small outward-opening non-return valve with a moderate spring load, the loading being insufficient to bring the nozzle within the closed nozzle classification but rather higher than is usual for the non-return valve of open nozzles. A second valve in the form of a flat plate is placed between the spring-loaded valve and pump as an additional safeguard against gas entering the system.

22. The Outward-opening Nozzle.

Nozzles having an outward-opening valve have not produced results which are anything like as satisfactory as those having the inward-opening differential valve. The outward-opening valve, owing to its large perimeter and the wide dispersal of the fuel resulting therefrom, always gives a soft spray, and although the fineness of atomization may be all that is required, such sprays tend to lack sufficient energy to carry the fuel far enough away from the nozzle to ensure the thorough searching of the combustion chamber. With quite small chambers satisfactory results have been obtained, but this type of nozzle offers considerably more difficulties in construction than does one fitted with a differential valve. Probably the worst fault of the outward-opening valve, and one which it seems to be impossible to eradicate entirely, is its tendency to give an unequal distribution of the fuel around its circumference. This is due to the valve lifting unevenly. Some clearance between the valve and its guide is essential, and this allows the valve to tilt slightly, giving a wider opening on one side than the other. The fact that the head of the valve is larger in diameter than the stem magnifies the effects of the tilt and aggravates matters. Valves designed to allow the stem itself to elongate and act as the spring and so avoid the use of sliding fits have failed to effect a cure. The trouble is probably due to a slight eccentricity between the stem and the surface of the valve against which the fuel pressure acts, causing the valve stem to bend or tilt instead of lifting squarely.

23. The Pintle Nozzle.

The annular orifice is not confined to the outward-opening valve and has found widespread and successful application in conjunction with the inward-opening differential valve. In this form, commonly known as the pintle nozzle, the nozzle is provided with a single hole of relatively large diameter, while the valve itself has an extension below the seat, the pintle, which projects down into the orifice, leaving only a small annular clearance between the pintle and the edge of the hole. The fuel escaping past the valve seat strikes against the pintle and is guided out through the annulus.

In its most usual form the pintle of this nozzle is not given a simple cylindrical form, but is shaped to give the fuel a cone of dispersal of the desired angle. The maximum cone angle obtainable varies with the diameter of the pintle, and at the present time is limited to about 60°. A typical example of a pintle nozzle is shown in fig. 149, which is a micro-photograph of an actual example after sectioning. The spray angle given by this particular sample is 30°. These nozzles really come under the heading of a variable-orifice nozzle, because the passage between the valve and its seat constitutes the major resistance. Apart

from this, the area of the annulus between the pintle and the hole in the body varies with the lift of the valve in certain cases. A characteristic of these nozzles is that, except under conditions of a very high rate of discharge, the valve oscillates at a high rate during injection. This is the normal behaviour of the nozzle, and the rate of discharge does not as a rule reach a figure such that the valve assumes a steady position at its maximum lift. The speed of oscillation can be varied by the strength of the spring. These nozzles give excellent service with certain types of chamber and have proved exceptionally reliable and trouble-free in service. The spray emanating from these nozzles is of the soft variety, and they are usually used with the spring set to allow the valve to open at a pressure around 100 atmospheres.

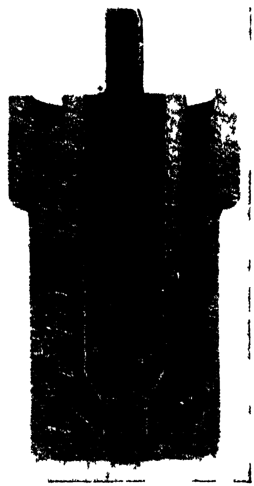


FIG 149—Section of pintle nozzle with valve at its maximum lift.

24. The Multi-hole Nozzle.

The multi-hole nozzle is a development of the single-hole nozzle with a heavily loaded differential valve. Instead, however, of a single hole of fairly large dimensions immediately beneath the valve and leading directly into the combustion chamber, a small sac is placed beneath the valve, and in this sac the desired number of holes are drilled, the holes being arranged to direct the fuel according to requirements. The sac may have one of several forms; it may be cylindrical or hemispherical, or may be made by a continuation of the cone forming the valve seat. In this latter instance the end of the valve is sometimes truncated in order to provide the necessary end clearance beneath it.

The number and arrangement of the holes can be varied at will, while the requisite length of hole is obtained by varying the thickness of the walls of the sac, and almost infinite variety can be obtained. If desired, holes of several different diameters can be provided.

The great drawback of nozzles of all types is carbon formation, which alters the distribution of the fuel. The carbon is produced by the decomposition of fuel in or on the nozzle as a result of heat, and one obvious means for avoiding carbon formation is to maintain the nozzle at a temperature low enough to avoid decomposition. This, however, is not by any means easy to achieve. In a high-speed engine the rate at which heat is liberated in the neighbourhood of the nozzle is exceedingly high and the cooling problem is acute. The provision of

special cooling arrangements is very difficult in the small space available, and even in large slow-speed engines it is sometimes difficult to accommodate such devices. In order to ensure a reasonable life the nozzles are made in alloy steels, which always have a low thermal conductivity, resulting in steep temperature gradients, which aggravate the cooling problem. As an alternative to improving the cooling of the nozzle, it has been suggested that if the temperature of the nozzle is allowed to exceed a certain minimum figure the carbon will be burned off and the nozzle kept free. This, however, leads to distortion and only a short useful life. Actually, distortion is very largely responsible for the trouble, or rather distortion accompanied by imperfect unloading, which results in a slow " weep " of fuel during the interval between injections and so promotes carbon formation.

Engines used for road transport work are subject to a condition peculiar to themselves. At intervals the engine is driven by instead of driving the vehicle, and under these conditions the governor cuts off the fuel supply entirely. The engine is still working with the full compression, and the nozzle is therefore subjected to the compression temperature without having the benefit of the cooling effect of the fuel passing through it, and in consequence reaches quite a high temperature. This temperature may not be so high as when running under load, but in all types of nozzles, except those with an outward-opening valve, a small quantity of fuel remains in the sac beneath the nozzle valve. Under power conditions this fuel is renewed by each injection, the fuel left in the sac forming the leading part of the next injection, but when the engine is being driven round instead of driving, this small body of fuel is subjected to prolonged heating and may break down and produce carbon deposits. This condition is particularly prevalent with the multi-hole nozzle, because not only is the quantity of fuel subject to this treatment rather greater than with other forms of nozzle, but the little sac, projecting as it does well into the hot gases, tends to reach a rather high temperature. The same thing applies, but to a less extent because the temperature is less, to the fuel immediately above the valve seat. Any small leak of gas into the system through an improperly seating valve will cause a serious aggravation of this trouble, and it is important to keep the volume of fuel at the nozzle down to the minimum.

CHAPTER X

Some Practical Results

1. The Compression Ratio Employed in Practice.

For reasons already given, it is not possible to give reliable information regarding the effect upon a given engine of a change of compression ratio alone. The range of compression ratio found in practice is a comparatively narrow one; at the lower end we are limited by the necessity for producing a temperature high enough to ensure ignition with a delay short enough to avoid rough running, while at the upper end the necessity for restricting the maximum pressure takes away any advantage that might otherwise have been obtained from increasing the ratio to a high figure. The low limit will be governed by the size of engine and the type of combustion chamber; a large engine can conveniently use a lower ratio than a small one because the larger mass of air results in a lesser loss of heat during compression, and although the mean temperature, as calculated from the compression pressure, may not be much higher than that produced with the same ratio and a smaller cylinder, the temperature at the centre of the mass of air will be considerably higher. At the upper end of the scale it has been shown (p. 68) that when the maximum pressure is limited to 1000 lb./sq. in. nothing is to be gained in economy by increasing the ratio beyond about 17 : 1. This figure was deduced from purely theoretical considerations for ideal conditions, only the loss occasioned by the increase in specific heat having been considered. Under practical conditions the loss of heat during combustion and the slowness of combustion will reduce the ratio to something less than 17:1.

Although there is nothing to be gained economically by increasing the ratio beyond a certain value, it does not follow that an advantage may not be gained in some other direction. The whole set-up of a compression-ignition engine is a compromise, and an increase in compression ratio will frequently mean a gain in all-round performance of the engine, even though there is no improvement from the thermodymanic standpoint. A higher ratio may result in a more compact chamber and one which is conducive to the retention or production of the necessary rate of swirl or type of air movement. It may also result in the chamber becoming better suited to the spray nozzle, and

so give improved results so far as the maximum smoke-free mean pressure is concerned. By reducing the delay period, and therefore the amount of uncontrolled burning, an increase in compression ratio may result in a decrease in the maximum pressure, while a decrease in the ratio may have the opposite effect.

In developing an engine of a new size or type there are so many variables that it is impossible to investigate the effects of all or even only a few of them, and it is therefore necessary to settle upon some features and leave these unaltered while adjustments and alterations are made to others which, for reasons generally associated with the manufacturers' individual facilities, are the ones most readily altered. To quote an example, with engines of the larger sizes and where the manufacturer makes his own sprayer nozzles it is usually cheaper and quicker to adapt the nozzle to the combustion chamber, whereas with the smaller engines running at high speeds the injection equipment is usually of a proprietary make, and although one starts with a selection of nozzle considered likely to cover requirements, variations are often much more readily made in the combustion chamber cavity than in the injection equipment. Much guidance is, of course, obtained from previous models, but information is not always capable of being scaled up or down.

To revert to the value of the compression ratio, the author is of opinion that from the point of view of fuel consumption per brake horse-power, there is very little improvement to be gained by increasing the ratio beyond about 14 : 1, but from the point of view of smoothness of operation, maximum smoke-free mean pressure, and speed range, there is a decided advantage in increasing the ratio to 16 : 1. The fact that a higher smoke-free mean pressure is obtained does involve an improved fuel consumption at the higher mean pressures, but this improvement is really due to improved combustion conditions and not to an improved cycle efficiency. The average service economy is not really affected; it is a case of the efficiency being maintained over a wider range of loads rather than a fundamental improvement in efficiency. That this seems to be the general experience is borne out by a published list * of particulars of high-speed engines. Of 502 models of compression-ignition engines for which the compression ratio was quoted, just under 70 per cent had a compression ratio from 15·0 to 18·0 : 1. The actual distribution was as shown in Table XXV.

It must be admitted that in a number of instances engines are included which are built up from a common cylinder with cylinders numbering from one upwards, and that the same engine adapted to a different service appears under more than one heading in some cases. This, however, is true of nearly all groups, and a particular instance occurs in the 13–13·9 group, where not only is there a large group

* *The Automobile Engineer*, June, 1938.

built up from a single size of cylinder but the same group appears twice owing to the original firm having licensed another manufacturer. Without detailed knowledge it is not possible to eliminate all such amplifications, and known cases have therefore not been removed from the list. An examination of the cylinder size in conjunction with the compression ratio does not entirely bear out the idea that the abnormally high ratios are associated with very small cylinders, although it is a fact that the lower ratios are associated with cylinders of comparatively large dimensions. The highest ratio given, 22 : 1, is associated with cylinders as large as 110 and 130 mm. diameter, but as the maximum pressure is said not to exceed 850 lb./sq. in., a figure less than would normally be reached during compression with a ratio of this magnitude, the correctness of this ratio may perhaps be questioned. The largest cylinders do not all have a lower ratio, cylinders as large as 230 mm. diameter being quoted as having a ratio as high as 16 : 1. The tremendous preponderance of engines having a ratio between 15 : 1 and 18 : 1 is certainly a significant fact, and indicates that in general the optimum results are to be obtained with a ratio between these limits.

TABLE XXV

Ratio	Number	Per Cent
Less than 13	26	5·2
13·0–13·9	55	10·9
14·0–14·9	57	11·3
15·0–15·9	100	19·9
16·0–16·9	104	20·7
17·0–17·9	107	21·3
18·0–18·9	37	7·3
19·0–22·0	16	3·2

2. Variation in Efficiency with Load.

In practice it is found that the variation in indicated efficiency with load, i.e. with the proportion of the available air utilized, follows the trend of the theoretical curve very closely, a steady increase in efficiency occurring as the load decreases. The results obtained from a number of engines having different types of combustion chamber are given in fig. 150, and show the improvement in indicated efficiency very clearly. The bulk of the results were obtained with a compression ratio between 14 and 16 : 1, but some results with a ratio of 12 : 1 given by H. B. Taylor * for maximum pressures of 600 and 800 lb./sq. in.

* *Proc. I. A. E.*, Vol. XXII.

are included and lie parallel to the other results. The figures given are for engine speeds varying from 1000 r.p.m. to 2000 r.p.m., and show that the efficiency actually attained is a high percentage of the figure theoretically possible after allowance has been made for the specific heats of the gases. Most of the figures refer to maximum pressures of between 800 and 1000 lb./sq. in. With 80 per cent of the air used an indicated efficiency as high as 39 to 42 per cent is shown, and at 40 per cent air used the figure has risen to 48 to 52 per cent. With constant volume working the cycle efficiency at these two points

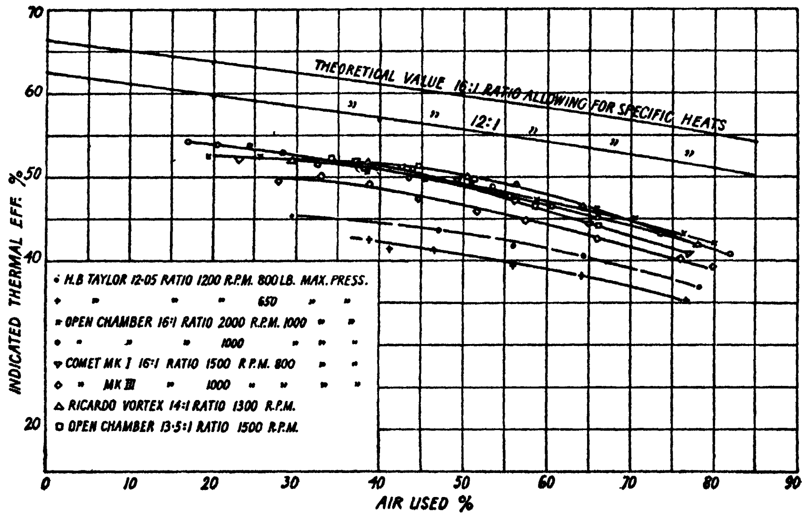


Fig. 150.—Variation in indicated thermal efficiency (on lower calorific value) with the quantity of air used

amounts to 55 per cent and a little under 61 per cent respectively, so that from 71 to 76 per cent of the attainable figure is being reached when 80 per cent of the air is used, and from 79 to 85 per cent when 40 per cent of the air is used. As most people are more familiar with such figures when expressed in terms of mean pressures, the same figures are shown in fig. 151 plotted against indicated mean pressures. The two sets of curves are similar in general form, although the positions of the curves are altered slightly in a horizontal direction by variations in volumetric efficiency. The figures show a remarkable degree of uniformity, and show also that under practical conditions there is not much to be gained in efficiency by increasing the compression ratio beyond 14 : 1, the results with ratios from 14 : 1 to 16 : 1 all lying together.

These results show that there is a very marked improvement in

efficiency as the maximum temperature is reduced by the reduction in the quantity of air used, the improvement, in round figures, amount-

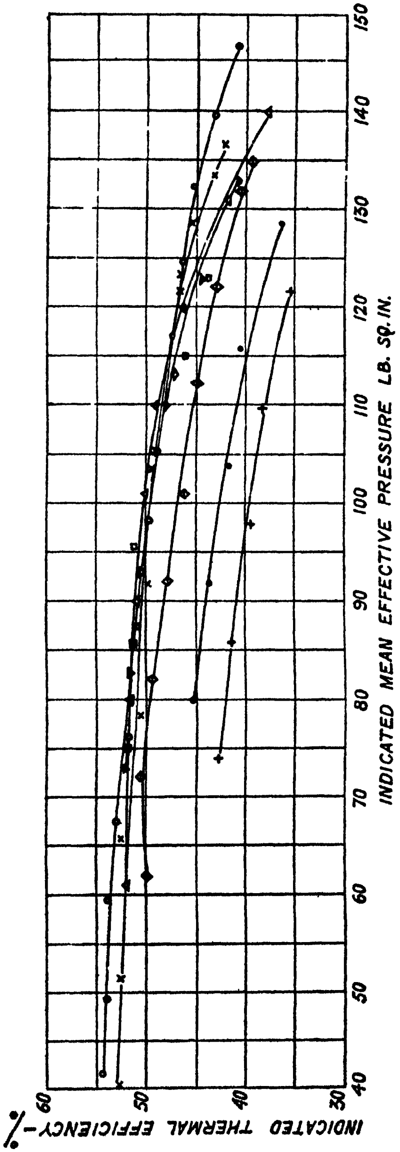


Fig. 151.—Variation in indicated thermal efficiency (on lower calorific value) with load

ing to an increase in the indicated efficiency from 40 per cent when 80 per cent of the air is burned to 50 per cent when only 40 per cent

of the air is burned, an improvement of 25 per cent based on the maximum output figure and one which goes far to offset the loss in mechanical efficiency at the lighter loads. The very real advantage gained from this improvement in the indicated efficiency is shown by Table XXVI, in which the brake thermal efficiency has been worked out for different loads on the assumption that a constant loss of 25 lb./sq. in. M.E.P. occurs at all loads and that a maximum indicated M.E.P. of 130 lb./sq. in. is developed at an indicated efficiency of 42 per cent. The brake thermal efficiency that would be obtained if the indicated thermal efficiency remained constant at 42 per cent has been worked out also, and the gain resulting from the improvement in indicated efficiency is shown.

TABLE XXVI

I.M.E.P. lb./sq. in.	B.M.E.P. lb./sq. in.	Efficiency Per Cent			
		Indic.	Brake	Brake for Indic. Constant at 42 per cent	Gain due to Increase in Indicated Efficiency per cent
50	25	52	26·0	21·0	24·0
60	35	52	30·3	24·5	23·9
70	45	52	33·4	27·0	23·6
80	55	51·5	35·4	28·9	22·6
90	65	51·0	36·8	30·3	21·6
100	75	50·0	36·6	30·7	19·2
110	85	48·0	37·1	32·5	14·1
120	95	46·0	36·4	33·25	9·4
130	105	42·0	33·9	33·9	—

These figures show very clearly that, in addition to the advantage gained from its ability to use a high ratio of expansion, the compression-ignition engine derives a tremendous further advantage at the lighter loads from the use of quality control. Under service conditions very few engines operate for more than comparatively short periods at or around their rated output; for by far the greater part of their time they are running at loads between about one-half to three-quarters of their rated output, and the advantage gained by the compression-ignition engine under these conditions is a very real one. The figures shown in Table XXVI may perhaps seem to be somewhat high, but they are given in terms of the lower calorific value, and, as will appear later, represent brake thermal efficiencies which are actually obtainable.

3. Mechanical Losses.

The direct measurement of the indicated output does not, as a rule, yield reliable results with high-speed engines. Pressure volume cards obtained from any form of indicator are far too subject to error to give even approximate results. Indicators giving pressure-time diagrams, while perhaps inherently more accurate, are subject to human maladjustment, and this can easily produce discrepancies at least as great as those of the other types. Quite a small error in the phasing of the recording drum will produce a disproportionately large error when the diagram is converted to a *pv* basis.

Many expedients have therefore been resorted to in order to obtain the indicated horse-power of high-speed engines, and while none of these can claim to yield perfect results, some at least yield results which may be taken as relatively correct when applied to engines of similar dimensions and speeds. Three methods are available, and with a reasonable degree of agreement between two the results may be accepted with a fair amount of confidence.

The first method was, the author believes, originated by Prof. W. Watson, and consists in cutting out each cylinder in turn and then measuring the loss in power after the speed has been restored to its original value. The lost power is assumed to be equal to the indicated horse-power of the cylinder not working. By doing this to each cylinder in turn the indicated horse-power of the complete engine is obtained. The method, which was originally employed for petrol engines, assumes that the mechanical losses of the inoperative cylinder are the same when the cylinder is idle as when it is running normally. If the readings are taken quickly, there does not seem to be any serious objection to this assumption, although there are two reasons for supposing that the readings may be slightly on the high side: (1) if the readings are not taken quickly enough, some decrease in the temperature of the oil in the cylinder walls may occur and cause an increase in piston friction because of a higher viscosity of the oil film; and (2) the exhaust gases, instead of being nearly all evacuated by their residual pressure as the exhaust valve opens, have to be forced out by the piston, a procedure which sometimes causes quite an appreciable rise in pressure during the later stages of the exhaust stroke. On the other hand, the piston ring friction is reduced when no ignition takes place, which acts as a compensating factor, and, on the whole, the assumptions made are fairly well justified. The method has the advantage that it can be employed with any form of dynamometer, but there is a difficulty which appears to crop up only when it is used for compression-ignition engines. At some speed, or rather over a range of speeds, readings are sometimes obtained which are out of line with those for the remainder of the speed range, sometimes on

the high side and sometimes on the low side. This is difficult to explain, but may perhaps be due to some alteration in the functioning of the fuel pump occasioned by one member having been put out of action. Readings taken by this method at a single speed must therefore be accepted with reserve unless confirmed by some other means.

The second method involves the use of an electric dynamometer powerful enough when used as a motor to drive the engine at the desired speed. The engine is run normally until the required temperature conditions are reached; the fuel is then cut off and the engine driven by the dynamometer, and the power absorbed is measured. This method makes the same assumption as the previous one, i.e. that the friction and pumping losses are the same when the engine is being motored as when it is running normally. One point requires care, namely, it will sometimes be found that the power required to motor the engine increases slightly during the first few seconds but finally settles down to a steady reading. This generally happens at the lower speeds and indicates an increase in friction during the first few seconds after the power is cut off. The differences may amount to as much as 5 or 6 lb./sq. in. of the piston area, and will be found to decrease as the load previously carried by the engine is reduced. To avoid this difficulty the reading should be taken as soon as a steady reading is possible, and to this end the dynamometer should be allowed to feed power back into the mains when the engine is firing. The speed can then be readjusted and the reading obtained in a minimum time after the fuel is cut off. Given suitable facilities, the reading can be taken very quickly and repeatable results can be obtained. The difference is probably due to changes in cylinder wall temperature and therefore in oil viscosity, assisted possibly by the contamination of the oil on the cylinder wall with fuel oil when the engine is firing.

The third method, which again makes the same assumption, is to take a series of readings at the desired speed but at different loads and to carry the readings right down to no load; if an electric dynamometer is available and can be made to feed power back into the mains, readings at indicated mean pressures of less than the no-load figure can be obtained. The results obtained are then plotted, either the horsepower or the brake mean pressure being plotted against either the fuel charge delivered per cycle, the fuel in lb. per hour or equivalent figures, or the percentage of the air used. The resulting curve is then extrapolated to zero fuel, or zero air used. The resulting negative value of horse-power or brake mean pressure is read off from the diagram and gives either the power required to turn the engine or the frictional resistance in terms of lb./sq. in. of piston area—usually referred to as the friction mean pressure.

This method was originally put forward by the author in a paper *

* *Proc. I. A. E.*, 1932.

read before the Institution of Automobile Engineers and other institutions, and has been found to give quite reasonable results. The readings almost invariably agree very well with those obtained by the motoring method and are applicable when a water brake or some other form of non-reversible dynamometer is used. The discovery that the reading obtained by motoring varies slightly with the power developed by the engine immediately before being motored suggests that this method will be subject to the same variation and that the reading obtained may be slightly greater than the correct reading for full power running. To be quite truthful, we must admit that there is no method which will give a really accurate measurement of the indicated horse-

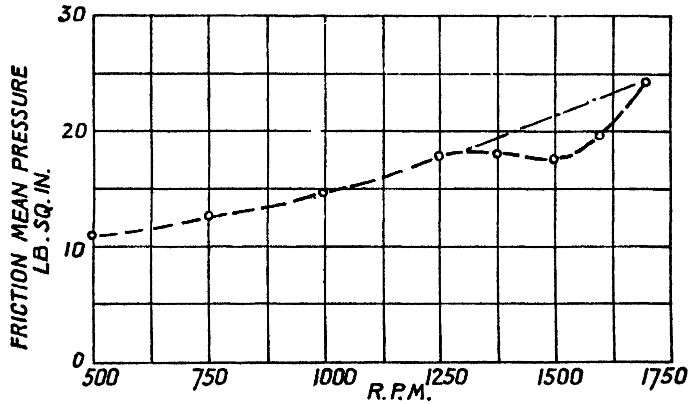


Fig. 152.—Showing inconsistency sometimes found when obtaining friction readings by Professor Watson's method

power developed by the engine. The discrepancy does not, however, appear to be serious; a difference of 3 or 4 lb./sq. in. in the friction loss between motoring the engine and the full load condition amounts to no more than between 2 and 3 per cent at the full load indicated M.E.P., and although we could wish the discrepancy could be avoided, the value obtained is probably at least as near the truth as are readings for slow-speed engines obtained from indicator cards.

Fig. 152 gives some readings obtained by cutting out each cylinder in turn and illustrates the peculiarity mentioned; at speeds around 1500 r.p.m. the friction loss apparently improves somewhat and is slightly less than the figure for 1250 r.p.m., a condition which is not confirmed by the fuel consumption given by the engine. The reading for 1750 r.p.m. falls on a fair curve with that from the lower speeds. Unfortunately readings at speeds above 1750 r.p.m. could not be obtained.

Fig. 153 gives readings obtained by motoring a single cylinder unit, and shows how at the lower speeds the friction loss increases as the load

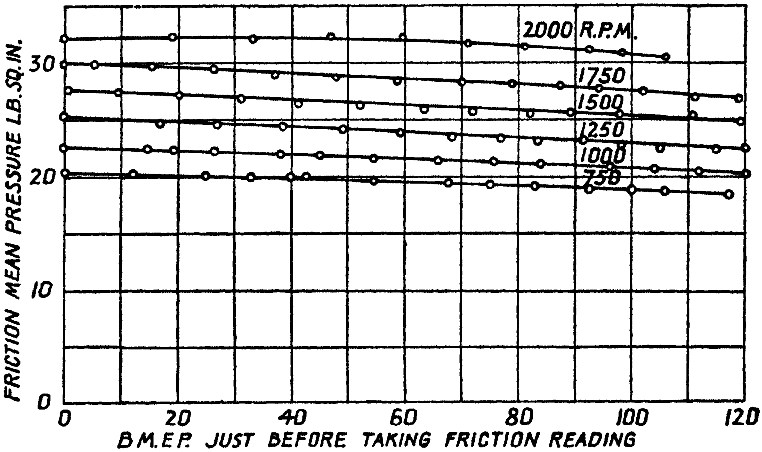


Fig. 153.—Variation in friction reading with load carried just before the reading is taken

decreases. The conditions under which these results were made were particularly favourable for taking readings quickly, and only a few moments elapsed between cutting off the fuel and getting the reading.

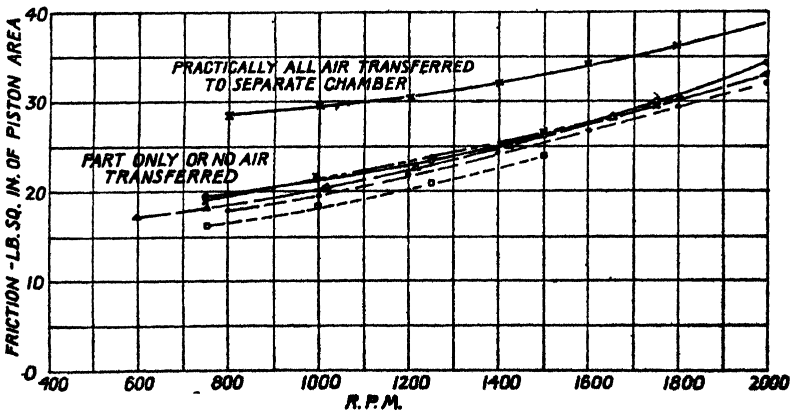


Fig. 154.—Typical friction loss curves obtained by motoring different makes of multi-cylinder engines

Prolonged motoring resulted in the friction loss rising to the maximum value for the speed under observation. In another instance it was found that at the higher speeds the change in load had no effect upon the friction loss and the friction mean pressure remained constant at speeds above 1250 r.p.m.

Fig. 153 also shows some readings obtained by the extrapolation method for the same engine as the one just quoted. These readings agree very well with those obtained under no load or prolonged motoring conditions, and where sufficient readings have been taken to enable a reliable extrapolation to be made the author has found the readings given by this method always to agree very well with those obtained by motoring.

Fig. 154 gives readings obtained from a number of engines of different types and sizes. All the figures were obtained by motoring, and are thus comparable with one another and represent typical values for the class of engine which is capable of speeds up to 2000 r.p.m. The influence of combustion chambers of different types is very clearly shown, those in which the air is transferred during compression to a separate chamber connected by only a small orifice to the cylinder showing an appreciably higher loss than engines in which no such transfer is made. The difference is due to two causes, the actual difference in pressure necessary to bring about the transfer of the air from one chamber to the other, and the increase in heat lost to the cooling water during compression, caused by the high velocity of the air over the surface of the combustion chamber.

4. Mechanical Efficiency.

The mechanical efficiency of the engine, although a matter of great importance, is a very misleading figure, and figures quoted from different engines are seldom truly comparable. This is because the value obtained depends so much upon the power developed, and an engine which in reality has a lower mechanical loss measured in absolute values may actually show a lower mechanical efficiency than an engine having greater losses. To quote an actual example, an engine which at a certain speed had a maximum smoke-free brake mean pressure of 90 lb./sq. in. had at the same time a mechanical loss of 24 lb./sq. in., giving a mechanical efficiency of 79 per cent, while at the same speed another engine of similar size had a maximum smoke-free brake mean pressure of 113 lb./sq. in. and a friction loss of 27 lb./sq. in., giving a mechanical efficiency of nearly 81 per cent. Compared at the same B.M.E.P., the relative values would have been 77 per cent for the second engine at the lower mean pressure and 82·5 per cent for the first at the higher mean pressure.

Instead of speaking in terms of mechanical efficiency, it is therefore much better to speak of the mechanical losses in terms of lb./sq. in. of piston area, this being a far better guide to the relative merits of different engines from the point of view of their friction losses than the mechanical efficiency, which will vary greatly according to the brake mean pressure.

5. Consumption and Efficiency on the B.H.P.

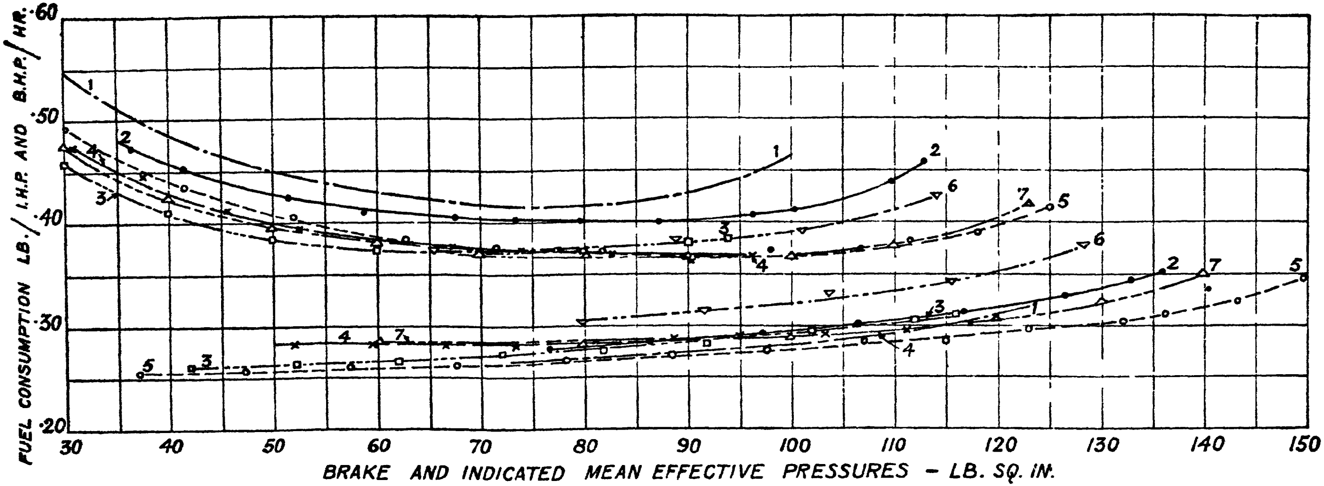
It may here be mentioned that with a given compression ratio and at a given degree of air utilization, the indicated efficiency obtained from different engines does not vary to any very great extent; in fact, the results obtained, even with quite wide variations in compression ratio, are extraordinarily consistent and uniform over the range of air utilization within which the air can be burned efficiently. Towards the upper limit of air utilization the efficiency falls off, but over the economic range the results from different engines vary but little (figs. 150, 151). Differences in friction losses will therefore show themselves in differences in the fuel consumption calculated on the brake horse-power, but at the same time it is the engines that have the relatively high mechanical loss on account of transferring the air from one chamber to another that are best capable of utilizing economically a high percentage of the air they receive, and are capable of retaining their combustion efficiency at high speeds. Some of the loss in consumption on the B.H.P. is thus compensated for by an improved performance in maximum brake horse-power and a wide speed range.

The effects of the friction losses upon the brake horse-power consumption are shown in fig. 155, which gives the consumption loops at 1500 r.p.m. for some of the engines for which the friction losses are given. The corresponding values of the I.H.P. are given also, and show how little is the difference between the results from different engines when they are compared on the I.H.P. basis. On the B.H.P. basis, however, there is a much wider variation. This is particularly noticeable with curves 1 and 2; both are from swirl-chamber engines, and upon the I.H.P. basis curve 2 is not quite so good as curve 1, whereas on the B.H.P. basis curve 2 shows a very material improvement over curve 1, the difference being entirely due to a modification to the combustion chamber, which resulted in a material reduction in the work done in transferring the air from the cylinder to the swirl chamber.

Curve 6 brings out the influence of the friction losses in a particularly striking manner. The figures shown are given by H. B. Taylor (*Proc. I. A. E.*, Vol. XXII); the engine had an unusually small friction loss, with the result that, although the consumption on the I.H.P. is noticeably higher than any of the other figures quoted, that on the B.H.P. compares very favourably with the best.

An engine with almost any form of separate chamber must suffer in that it has higher mechanical losses than engines with an open chamber, but its ability to use a high percentage of the air it receives is due to the high velocity air movement which it is capable of producing and which is difficult to obtain by other means.

The curves in fig. 155 show fuel consumptions of the order of 0·37 to 0·38 lb./B.H.P./hour; such figures, and even better ones, are



1. 8·8 LITRE COMET I 2. 8·8 LITRE COMET III. 3. 6·9 LITRE OPEN CHAMBER 4. 8·8 LITRE OPEN CHAMBER
5. SINGLE CYL. 1·65 LITRE OPEN CHAMBER. 6. H.B. TAYLOR (PROC I.A.E. VOL. XXII) 7. RICARDO "VORTEX" CHAMBER

Fig. 155.—Fuel consumption on B.H.P. and I.H.P. of representative types of engine

becoming increasingly more common in this country, although it must be admitted that figures of 0·40 lb./B.H.P./hour and upwards are far more general if engines the world over are considered. The lower consumptions are associated with the open type of chamber and are generally accompanied by limitations of the maximum smoke-free brake mean pressures, or the maximum speed at which a consumption approaching the optimum can be obtained, or both. The heavy taxation on fuels for road transport in this country has resulted in focusing attention upon fuel consumption, and of recent years the better

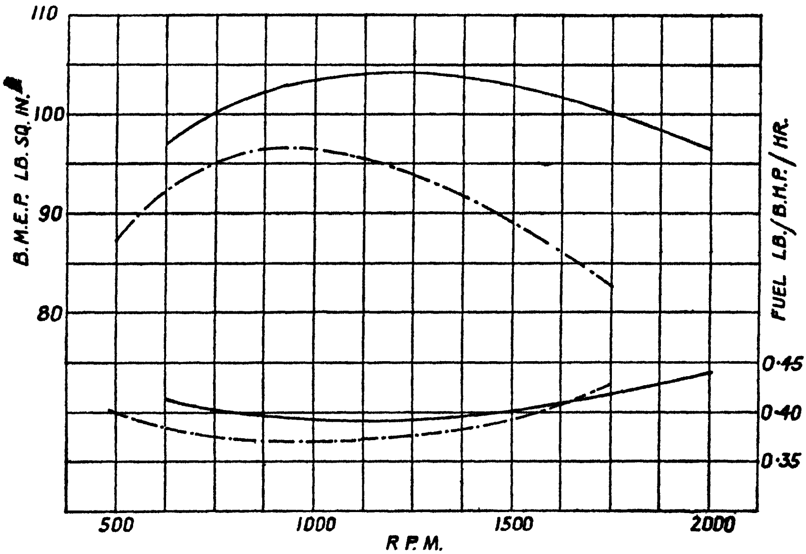


Fig. 156.—Comparative performance of an open combustion chamber and a swirl chamber engine; full line swirl chamber, chain line open chamber

consumption of the open chamber has been sufficient in many instances to outweigh its shortcomings in other directions. Progress is, however, being made with both types of chamber, and improved consumptions are being obtained by the indirect methods without making any sacrifice in the maximum B.M.E.P. or the speed range. The improved brake mean pressures are being obtained as a result of reduced mechanical losses, while with the open combustion chamber engines improved brake mean pressures are being obtained at the higher speeds by better combustion conditions, with a consequent improvement in fuel consumption. Generally speaking, there is a tendency for the open chamber to have a better consumption than the other type at speeds below 1500 r.p.m., while at speeds above this figure the indirect methods—or at least some indirect methods—show the better results.

The higher efficiency attained by the open chamber should allow it to reach a higher brake mean pressure than is reached with the indirect chamber, and the fact that it does not invariably do so is due to its inability to utilize so great a proportion of the air. Usually the open chamber is unable to utilize more than about 60 to 65 per cent of its air supply, whereas the indirect chamber can economically utilize 80 per cent, sometimes even more. At the same efficiency this would mean that the indirect chamber would develop one-third more indicated horse-power for the same cylinder capacity, and differences of this order are commonly recorded. The energy expended in bringing this additional quantity of air into service, however, reduces the margin of output in terms of the brake horse-power and sometimes results in there being little difference between the two types of chamber over the more moderate speeds of 750 to 1250 r.p.m. At higher speeds the ability of the open chamber to consume air is apt to fall off and the brake horse-power suffers, whereas the indirect chamber frequently shows no falling off in its ability to make use of a large portion of the air, and, although the mechanical losses increase with the speed, they do not do so at such a rate as to equal the falling-off of the open chamber, with the result that power is better maintained at high speeds. This is shown by fig. 156, which gives the B.M.E.P. and consumption on the B.H.P. for a representative member of each type.

6. Heat Balance.

An accurate balance accounting for the whole of the heat appears almost impossible of attainment with an internal-combustion engine, and balances usually include a convenient item "radiation, &c." or "unaccounted for". Ultimately there is only one item that really counts: the heat converted to useful work. How the remainder is disposed of concerns the engine user on one count only, namely, the heat carried away by the cooling water. This has to be taken care of in some way, and the quantity of heat thus disposed of governs the extent of the provision that has to be made either in the size of the cooler, or radiator, or in the quantity of cooling water.

The heat converted to useful work, brake horse-power, amounts, according to the consumption obtained, to 33 to 37 per cent of the heat supplied. On the indicated horse-power basis the figures run from 45 to 50 per cent under full-load conditions, depending upon the maximum quantity of air the engine is capable of using.

The total quantity of heat appearing in the cooling water depends upon a number of factors. The extent of the air movement during combustion and the surface exposed to the hot gases are the major factors, while the amount of water-jacketed exhaust port plays a not inconsiderable part. The rate of water circulation is yet another factor. In any given engine the ratio of the heat carried away in the

cooling water and the heat converted into work tends to decrease as the speed increases, but approaches a more or less constant value towards the upper end of the speed range, i.e. the heat lost per B.H.P. hour decreases with speed.[4] This decrease in the proportion of the heat carried away by the cooling water does not materially influence the efficiency of the engine, because only a fraction of the total quantity of heat represents an actual loss of efficiency. There is probably some reduction in the true heat loss as the engine speed increases, and any such reduction should be accompanied by an improvement in the indicated efficiency. Instances can be found of an improvement in the indicated efficiency with an increase in speed. The probability of some improvement in combustion efficiency being, in part at least, responsible for such an increase in efficiency must not be overlooked; at the same time, the fact that no such improvement is found cannot be accepted as positive evidence that no change in the true heat loss has taken place. It is, of course, quite possible that the whole of any change in the heat to water may be due to an exchange between the exhaust loss and the jacket loss. That some such exchange is taking place is indicated by the fact that, for a given degree of air utilization, the temperature of the exhaust gases increases with an increase in engine speed. The author has found that on air-cooled engines the heat carried away in the cooling air can be varied over wide limits without any change in the heat converted into work. Fundamentally, such a condition must apply equally to water-cooled engines, although it is not so readily detected because, with the initially lower cylinder wall temperature of the water-cooled engine, the reduction in jacket temperature necessary to induce the higher jacket loss will cause an increase in piston friction by reason of the increased viscosity of the lubricating oil on the cylinder walls.

Some typical jacket loss figures are given in fig. 157, which shows the heat loss expressed as a ratio, Jacket loss/Heat into work, for several different types of engine; the much lower rate of heat loss to water for engines with an air movement of a lower order is clearly shown.

At any one speed, the jacket loss, expressed as a proportion of the total heat supplied, does not vary much with the load but remains fairly constant over a wide range of loads. It does, however, show a falling off at the lowest loads. This is the logical tendency; the indicated efficiency increases as the load decreases, the sum of all the losses must therefore be reduced, and it is only natural that the proportion of heat going to the cooling water should be reduced also.

The total quantity of heat carried away in the cooling water under full-load conditions varies from around 35 per cent to as low as 25 per cent. Of this quantity some must be considered as representing part of the mechanical losses of the engine. Piston and piston ring friction will appear as heat in the cooling water, while the heat lost during

compression and representing work lost and not returned during re-expansion is a part of the motoring losses of the engine and appears as heat in the cooling water These two together may represent a not inconsiderable proportion of the total heat in the cooling water, and

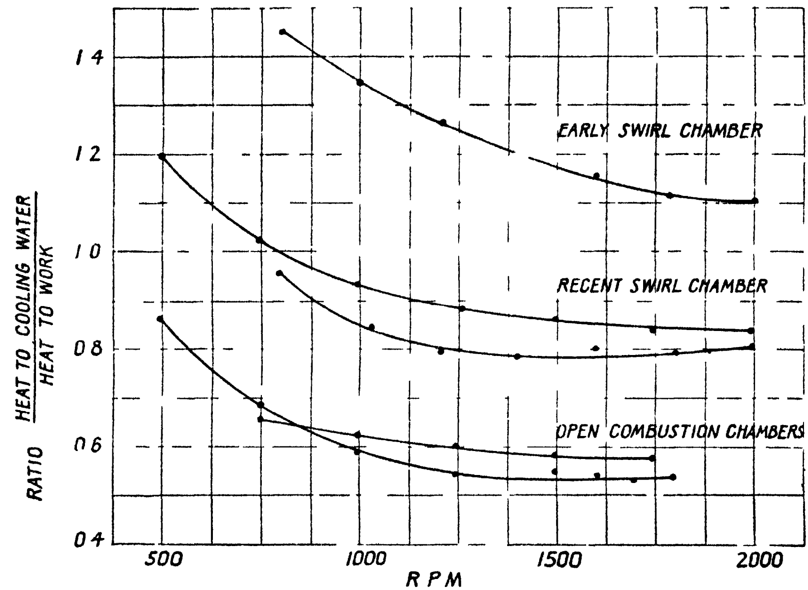


Fig 157 —Typical jacket losses for different types of combustion chamber

the author has known instances where they represent as much as 18 per cent of the total, as determined by a motoring test at a speed of 1500 r.p m.

7. Comparison of Results from different Engines.

On hearing results quoted from outside sources, one is frequently anxious to know how these figures compare with one's own experience. The figures most commonly given are the specific fuel consumption and the corresponding mean pressure. These two figures, while they give the information most interesting from a commercial standpoint, do not immediately give much idea of the real effectiveness of the engine as a potential power-producing unit. Engines developing a given mean pressure may do so with rather widely differing consumptions, and the engine giving the lowest consumption is certainly the best machine thermodynamically. If, however, the high efficiency is attained by making very poor use of the air, it is quite possible that an

engine which does not have quite so good a consumption may be the better machine from an all-round commercial standpoint.

Given the brake, or indicated, mean pressure and the corresponding consumption, it is quite a simple matter, by working out the energy supplied per cubic inch of cylinder volume, to asses with a fair degree of accuracy the extent to which the engine is making use of its air. From the usual horse-power formula, the horse-power developed per cubic inch of cylinder volume swept per second is given by

$$\text{H.P./c. in. sec.} = \frac{P_m \times 1 \times 60}{33{,}000 \times 12 \times 2} = \frac{P_m}{13{,}200},$$

where P_m is the mean pressure developed in lb./sq. in.

Taking the calorific value of the fuel as 10,000 lb. calories per lb., the energy supplied per cubic inch of cylinder volume will be

$$\frac{w \times \text{H.P.} \times 10{,}000 \times 1400 \times 2}{3600 \times 1728} \text{ ft.-lb./c. in.}$$

for a four-stroke engine, where w is the weight of fuel consumed in lb./H.P.-hour.

If we substitute in this equation the value for the horse-power just determined, the expression reduces to

$$\frac{w \times P_m \times 10{,}000 \times 1400 \times 2}{3600 \times 13{,}200} = \frac{w \times P_m}{1{\cdot}7} \text{ ft.-lb./c. in.}$$

It has been shown elsewhere (p. 36) that the maximum amount of energy liberated when the whole of the available air is used amounts to 45 ft.-lb./std. c. in., and it has been shown also (p. 89) that the average absolute volumetric efficiency under operating conditions may be taken as 75 per cent. The average maximum energy per cubic inch of cylinder volume when all the air is used is therefore $0{\cdot}75 \times 45 = 33{\cdot}7$ ft.-lb., and the air utilization of the engine is therefore equivalent to

$$100 \times \frac{w \times P_m}{1{\cdot}7 \times 33{\cdot}7} = \frac{w \times P_m}{{\cdot}57} \text{ per cent.}$$

As the volumetric efficiency is a somewhat variable quantity, however, the expression may reasonably be taken as

$$\text{Percentage of air used} = \frac{w \times P_m}{{\cdot}57} = 1{\cdot}75\, w \times P_m.$$

If there is reason to believe that the volumetric efficiency differs materially from the figure of 75 per cent, the constant must be varied

accordingly, but with the constant given above the result obtained will be sufficiently accurate for general purposes.

This expression is of course applicable to normally aspirated engines only.

8. The Air used by Actual Engines.

It is a matter of real importance that engines should be compared as to their relative ability to use up the air they receive. In the case of the smaller fast-running engines, such as those used for road transport work and some of those used for marine work, the petrol engine is the standard by which their performance is judged. These latter commonly have a brake mean pressure in the neighbourhood of 100 lb./sq. in., and there has been a tendency in the past to look upon this mean pressure as the figure to be aimed at for the compression-ignition engine. Up to a point this is correct; in order not to be handicapped the compression-ignition engine should not fall short of its competitor in specific output, especially as it is inevitably somewhat heavier, but it is certainly not right to rest content with this figure. The higher efficiency attainable with the compression-ignition engine entitles us to expect a higher mean pressure than is obtainable from a petrol engine, and the failure to achieve a higher mean pressure is indicative of inefficiency in other directions, e.g. air utilization. In the case of larger engines the stresses produced by the increased temperatures associated with a high specific output necessitate some limitation in the quantity of air burned, but with engines designed to compete with the petrol engine this condition should not arise. The lower efficiency of the petrol engine, coupled with its natural ability to utilize the whole of the available oxygen, results in much higher temperatures than are found in compression-ignition engines, and this being so, there is no reason why the reliability of the latter should suffer therefrom. In certain instances the sprayer nozzle presents difficulties arising from the accumulation of carbon if the mean pressure is increased beyond a certain figure, but to accept this as a hard-and-fast limitation is to place a bar on all progress. It may be necessary to reduce the output from an engine somewhat for the time being, but this is a very different matter from accepting the condition as a positive limiting factor. If we look back over the difficulties which in the past appeared to be insuperable and which now are no longer considered even to be difficulties, we must surely feel that such a condition must not be allowed to limit engine development. Theoretically a low fuel consumption should be associated with a high mean pressure, but in actual practice the reverse is commonly the case, a really low consumption going with a medium or relatively low mean pressure and a high mean pressure going with a relatively high consumption. For any given mean pressure, the lower the fuel consump-

tion the lower is the air utilization factor. This is very clearly shown by the expression connecting consumption and mean pressure with air utilization as given above:

$$\text{Percentage of air used} = 1{\cdot}75\, w \times P_m.$$

This means that for a given mean pressure the air utilization efficiency of the engine decreases directly as the fuel consumption improves. Actually the converse should be the case, and any improvement in consumption should be associated with the same degree of air utilization and therefore result in an increase in the mean pressure; the air utilization efficiency should not have to suffer if we want a low consumption, as is now commonly the case. It unfortunately happens that devices intended to assist in the utilization of a large proportion of the air often have the result in decreasing the mechanical efficiency and thus offsetting some of the gain.

This does not seem to be really necessary, and a careful study (admittedly a difficult problem) of the air movement during the combustion period should do much to help matters. The answer seems to lie in the better co-ordination of fuel and air movement; this is indicated by the difference in the form of the consumption loops of the open chamber engines using only a moderate proportion of the air and those for the indirect type using a large proportion of the air. In the former, the exhaust begins to colour up and gets gradually dirtier over a wide range of loads without any marked falling-off in efficiency, whereas in the indirect type with a vigorous swirl the consumption loop generally turns upwards somewhat before the exhaust begins to show colour and the exhaust then gets rapidly dirtier, with a sudden rise in consumption, becoming very black after only a quite small further increase in M.E.P. This increase in consumption before the exhaust colours up is due to the decrease in efficiency caused by the higher temperature associated with a high degree of air utilization and the rapid rise in consumption, and blackening of the exhaust is due to the difficulty in finding the oxygen during the later stages of a high degree of air utilization. The utilization of 80 per cent of the oxygen before the exhaust shows any colour is possible with a high swirl engine, whereas 60 per cent is a more usual limit for the open type of chamber. If 80 per cent of the oxygen can be used without colour in the exhaust with one type of chamber, there is no fundamental reason why it cannot be done with another; that it is not done is due to the faulty distribution of the fuel and the air, resulting in some part of the air receiving an excess of fuel, an excess which increases steadily as the total quantity of fuel injected is increased.

That it is possible to produce a B.M.E.P. of the order of that which the high brake thermal efficiency of the open chamber suggests is obtainable is shown by curves 5 and 7 in fig. 155. Curve 5 was obtained

from a single-cylinder experimental engine, 120 mm. bore, 146 mm. stroke, running at 1500 r.p.m. in the A.E.C. laboratory. The engine was fitted with a chamber similar to that in fig. 105*b*, p. 206; the maximum B.M.E.P. at the above speed was 124·5 lb./sq. in. with a just visible exhaust at a consumption of 0·413 lb. fuel/B.H.P.-hour, and represented an air utilization of 86·6 per cent by actual measurement. Curve 7 is from figures obtained by Ricardo from one of his sleeve-valve engines, also a single-cylinder machine, 5½ mm. bore × 7 in. stroke, running at 1300 r.p.m., and at a B.M.E.P. of 123 lb./sq. in. on a consumption of ·42 lb./B.H.P.-hour, representing an air utilization, also measured, of 89·6 per cent.

Unfortunately figures such as these do not yet seem to be possible with multi-cylinder units, but the fact that they can be obtained on single-cylinder experimental units indicates that such results are actually possible and that careful study of the movements of the air and the fuel should lead to such results being obtained in everyday service. The reason for the discrepancy between single-cylinder and multi-cylinder results is difficult to understand in the case of compression-ignition engines. There is no question of distribution such as occurs with petrol engines, and the answer seems to lie in some interference to the air movement produced by the additional cylinders; that some such influence exists is shown by the fact that a slightly different position has been found necessary for the mask on the inlet valve of the multi-cylinder engine as compared with that found to give the optimum results on the corresponding single-cylinder unit. If we can obtain results such as the above without resorting to means that either increase the mechanical losses of the engine or reduce its breathing capacity, then brake mean pressures of 130 lb./sq. in. and upwards on a consumption of the order of 0·40 lb./B.H.P.-hour should not be impossible from normally aspirated engines.

In order to illustrate the variations in the quantity of air used by different engines the values given in fig. 155 are shown in fig. 158, plotted on the basis of air used; as the actual values of air used are not available for every case, the expression given on p. 310 has been used where the actual values are not available.

9. The Duration of Combustion and the Temperature reached.

The length of time which elapses before combustion can be considered to be complete is a matter of considerable importance, because the efficiency of the engine is directly influenced thereby. In this connexion it is of interest to note that, despite the fact that the fuel and the air are kept separated until combustion should begin, the combustion in a compression-ignition engine is completed in a much shorter period of time, and therefore occupies a much smaller fraction of the whole cycle, than that in a petrol engine of similar size running at the

same speed. Despite the existence of the delay period, the beginning of injection of a compression-ignition engine is timed to take place much later than the ignition of a petrol engine, but even so the combustion is completed much earlier in the expansion stroke. This

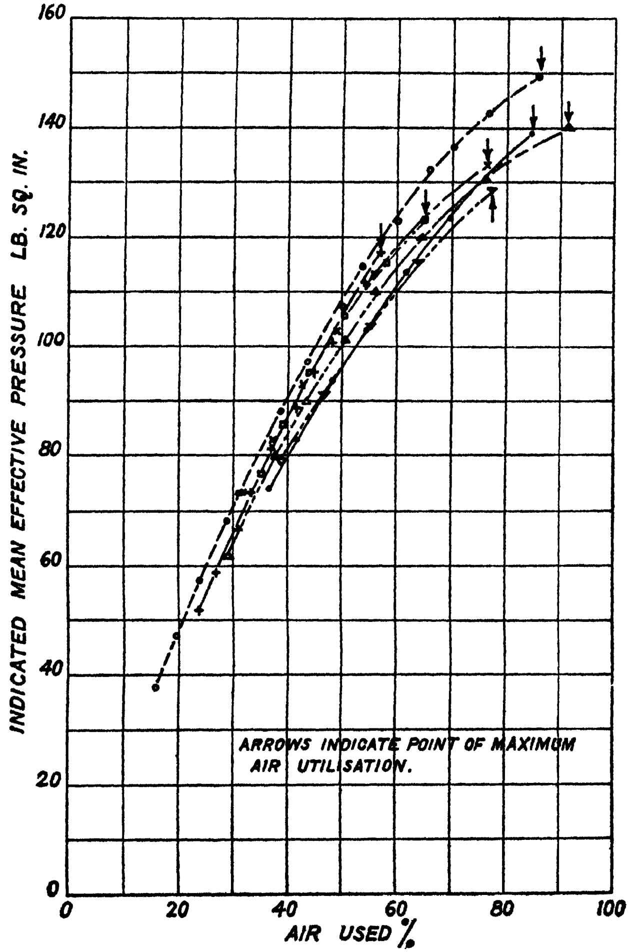


Fig. 158.—Variation in I.M.E.P. with air used and maximum quantity of air used by different engines

condition is brought about by the much greater pressure and density under which combustion takes place, the vigorous nature of the air movement and the higher temperature at which combustion begins, all of which help to speed up the process to an extent which cannot be approached in a petrol engine without producing serious trouble from detonation.

The duration of combustion can be ascertained with a reasonable degree of accuracy if suitable indicator diagrams are available. The author does not share the faith which is placed in indicator diagrams by some branches of the engineering profession, but he does consider that, when used with discretion and a full realization of their shortcomings, indicator diagrams can be made to yield much valuable information on the conditions within the cylinder of a high-speed engine. Such information must, however, be looked upon as being of a relative nature rather than as giving absolute values. For the purpose of determining the point at which combustion is completed, the Farnboro' indicator is probably the most suitable, as it gives good large diagrams and is easily understood and handled by those without any specialized knowledge.

To ascertain the duration of combustion the pressures on the absolute scale must be plotted upon a log-log basis against the corresponding volumes or expansion ratio figures; the point at which the adiabatic expansion begins can then be detected quite easily, and in some instances a sharp change in direction takes place at the termination of combustion and the beginning of adiabatic expansion.

From diagrams suitable for determining the duration of combustion the temperature reached at various points throughout the cycle can be determined with a fair degree of accuracy. If two points are taken, one on each side of, and equidistant from, top dead centre, and the pressure at each of these points is measured upon the absolute scale, then since the volumes at the two points are equal, the absolute temperature at the two points will bear the same ratio to one another as do the absolute pressures, i.e. $T_2/T_1 = P_2/P_1$. The temperature at any point on the compression stroke can be calculated with a reasonable degree of accuracy, especially if measurements of the volumetric efficiency and of the exhaust temperature are available, and the temperature of the corresponding point on the expansion stroke can therefore be determined as described above. By taking a number of points close together near the top dead centre a very fair idea of what is happening during the combustion period can be gained. The temperature figures together with the corresponding volumes can be plotted on the log-log basis and used to determine the duration of combustion, although the adiabatic part of the temperature curve, not making such a steep angle with the part covering the combustion period, will not give so sharp a break as that obtained if the pressures are plotted.

For determining the point at which the combustion may be considered to be complete, this method may be considered as satisfactory, but from the point of view of ascertaining the temperature reached during combustion it is open to the objection that the volume of the gases has been increased by combustion, and the calculated tempera-

tures are therefore correspondingly greater than those actually reached. With the assumption that not more than 80 per cent of the available air is burned, the maximum discrepancy from this cause will be around 5 per cent, and will attain this magnitude only from the point at which combustion is completed onwards. Between this point and the commencement of combustion the discrepancy will vary according to the degree of completion attained by the combustion, and any correction here is difficult to make because of the difficulty in ascertaining the exact progress of combustion at any particular point. Further, we do not know with any exactitude what really happens during combustion; all we know with any degree of certainty is the ultimate outcome of the combustion. By taking equilibrium conditions into consideration it would be possible to arrive at an approximation to the state of combustion at any given point, but this is certainly too great an elaboration to undertake in connexion with any ordinary engineering development, and, if correction is felt to be necessary, the simplest thing to do is to correct the temperature for points after the completion of combustion and then to graduate the correction from the maximum figure at the end of combustion down to zero at the beginning. For most purposes it is probably sufficient to ignore the correction altogether; the maximum error is of the order of 5 per cent, and it is doubtful whether the accuracy of the whole method is any closer than this.

The measurement of the pressures at the critical part of the diagram is greatly facilitated if the indicator drum is driven at twice the speed of the crankshaft, as by this means the slope of the pressure rise is halved and the accuracy of measurement improved. The accuracy of the whole method depends very largely upon the correctness of the phasing of the indicator; any fault in this will lead to a disproportionately large discrepancy in the results, but if the phasing is done with reasonable care the results will not be too seriously affected.

A point to be remembered is that the temperature obtained by this method, or by any other that is based upon pressure measurements, gives the mean temperature of the whole mass of gas. The conditions under which combustion takes place in a compression-ignition engine are such that there must be wide variations in temperature at different points of the combustion space, just as there are at various points in the combustion chamber of an oil-fired furnace. This will be particularly marked at the lighter loads when only a small portion of the air is burned and combustion is taking place in a comparatively restricted zone, or zones, in line with the orifice, or orifices, of the sprayer nozzle. This being so, the question arises whether the maximum local temperature may not be nearly the same under all conditions of load and whether it may not reach figures approaching those reached in a petrol engine. This idea raises the question whether

one is really justified in assuming that dissociation can be ignored in the compression-ignition engine. The fact that such a zone of intense temperature is surrounded by gases containing a large excess of oxygen should, however, render any dissociation of exceedingly short duration. Further, as any allowance made for dissociation would reduce the theoretical efficiency of the cycle, the figures actually being obtained in service would then be placed in an even better light than if dissociation were ignored; but one hesitates to lower the standard by which results are measured.

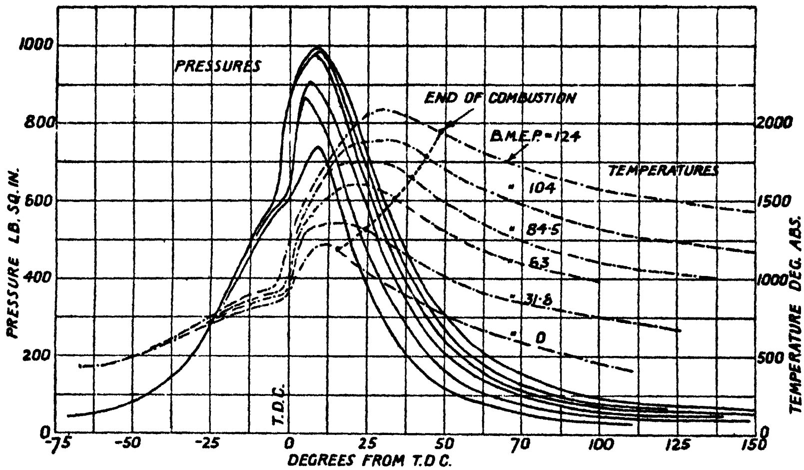


Fig. 159.—Pressures and temperatures at different loads in an open combustion chamber at 1500 r.p.m. Compression ratio 16 : 1

Some indicator diagrams examined by the method just outlined are shown in figs. 159 and 160. The diagrams were taken at varying loads, with the engine, an open combustion chamber design, running at 1500 r.p.m. Fig. 159 shows the original diagrams and the temperatures plotted against a base of crankshaft rotation, and fig. 160 shows these plotted logarithmically against the expansion ratio, the volume at top dead centre being taken as unity. At the highest load, which corresponds to the utilization of 64 per cent of the air, the expansion ratio up to the point at which combustion became complete is as high as 4·5 : 1, corresponding to a crankshaft rotation of about 50°; as the load is reduced, however, the combustion is completed earlier and earlier in the cycle, and by increasing the effective expansion ratio brings the efficiency nearer to the ideal figure. These diagrams indicate that the value of n during expansion is about 1·35 as compared with 1·28 for petrol engines, and show how much earlier in the expansion stroke the bulk of the heat has been received, and also the lower mean specific heat of the gases. The maximum temperatures reached are of

a fairly high order when it is considered that so small a proportion of the available air is used; this, however, is accounted for by the fact that combustion starts from a temperature of nearly 600° C.

The relative rates of combustion at different engine speeds may be

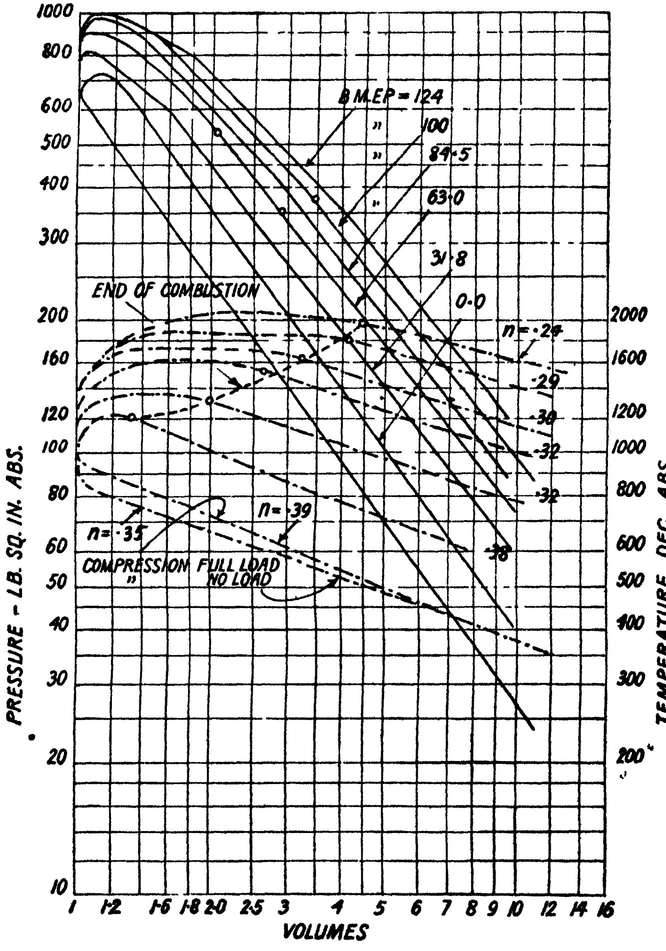


Fig. 160.—PV and PT diagrams given in fig. 159 plotted upon a log-log basis

obtained by comparing diagrams taken at different speeds for the same weight of fuel per cycle, as is shown in fig. 161.

Provided the work has been carefully done and good diagrams have been obtained, the results obtained can be very instructive, but too much must not be expected. Some trouble is experienced with points coming wide of the general run of the readings. This may usually

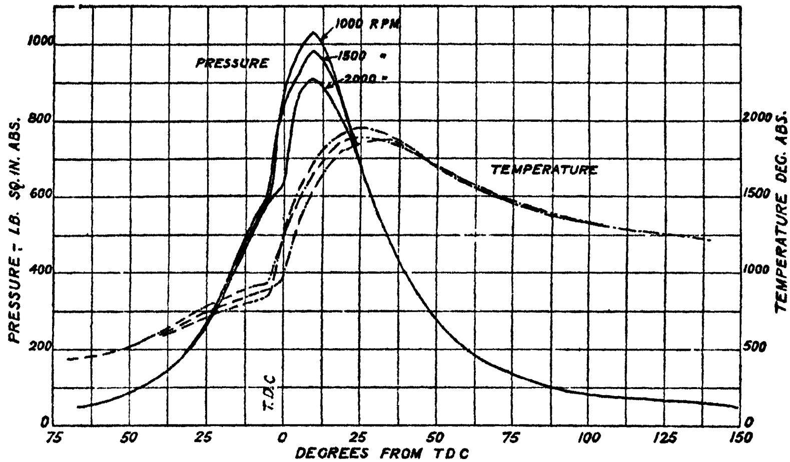


Fig. 161.—Pressures and temperatures with substantially equal fuel charges at 1000 1500 and 2000 r.p.m. Open chamber, compression ratio 16 : 1

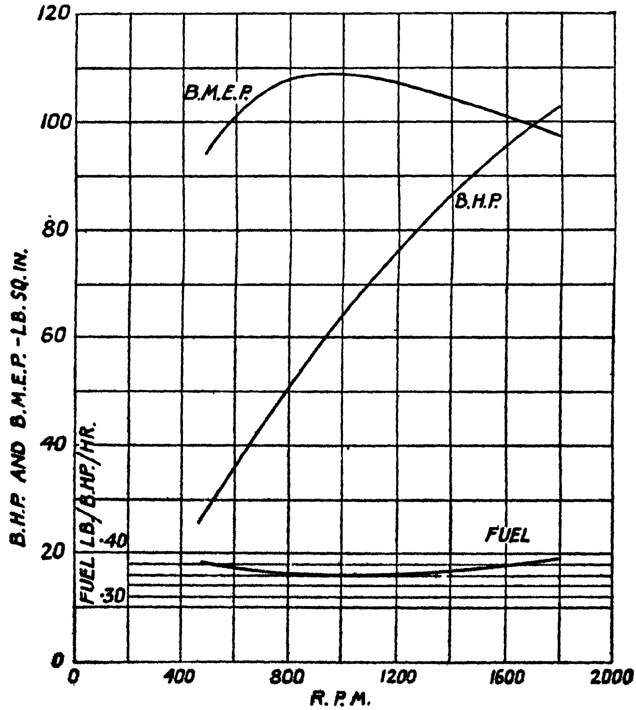


Fig. 162.—Performance of a 6-cylinder engine of 7·7 litres capacity with an open combustion chamber of the toroidal type

be traced to pressure waves in the indicator passage; it is not always possible to make this passage as short as it should be, and at the same time the passage in the Farnboro' cylinder unit is itself of some length. This latter may be avoided by dispensing with the water-cooled cock

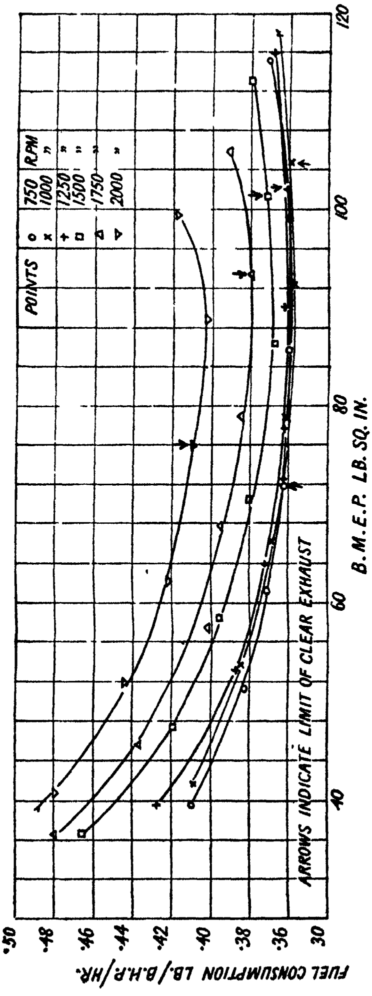


Fig. 163.—Consumption loops at several speeds for an engine having an open combustion chamber of the toroidal type

and replacing it with a special short connexion. Water jacketing of the cylinder unit is quite unnecessary for compression-ignition engine work, and for special work the elimination of the cock is no serious disadvantage, although it shortens the life of the indicator valve before reconditioning is required.

In order to illustrate the all-round performance of the modern high-speed engine, the results from two up-to-date vehicle engines are given in figs. 162 to 165. One of these engines has an open combustion chamber of the "toroidal" form (fig. 105, p. 206), while the other, a slightly larger engine, has a Ricardo "Comet" Mark III chamber (fig. 112, p. 217). Fig. 162 shows the speed-power curve together with the corresponding information as to mean pressures and fuel consumption of the open combustion chamber engine, while fig. 163 shows the

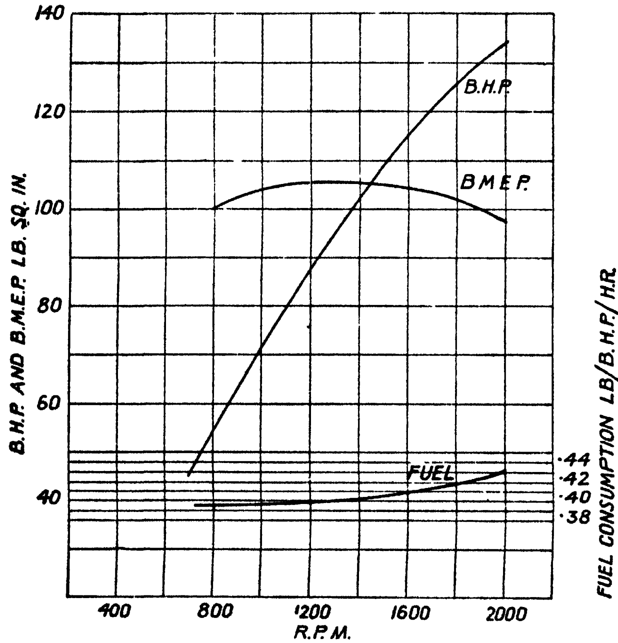


Fig. 164.—Showing performance of an 8·85 litre engine fitted with Ricardo "Comet" Mark III combustion chamber

load-consumption loops for the same engine at a number of different speeds. Figs. 164 and 165 give the corresponding information for the engine with the "Comet" chamber. These two sets of curves serve to bring out the relative merits of good representatives of the two types. The much higher B.M.E.P. obtainable from the swirl chamber at the clear exhaust limit, especially at the higher speeds, and the engine's ability to operate at high speeds without smoke are well brought out, as is also the fact that the smoke limit is not reached until the consumption loop shows signs of a rapid rise in consumption owing to the lack of readily available oxygen.

The open combustion chamber figures show the somewhat lower

consumption figures which this type is capable of giving, and also show how the limit of the clear exhaust occurs at a point well below that at which the consumption loops show any tendency to turn sharply up-

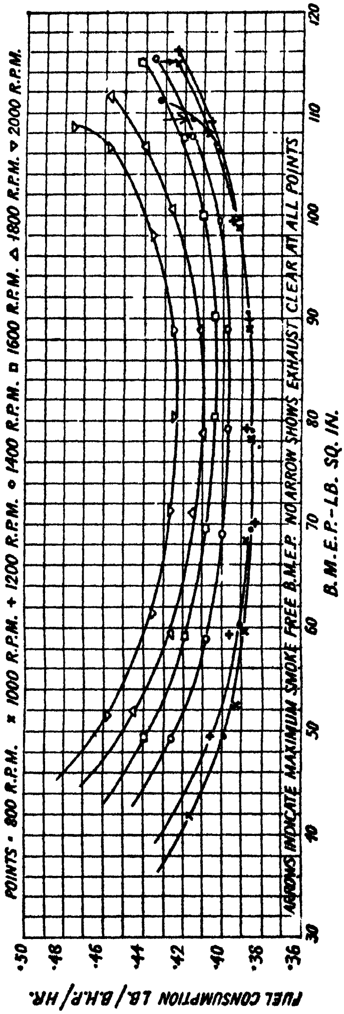


Fig. 165.—Showing fuel consumption loops at several speeds for an 8·8 litre engine with "Comet" Mark III combustion chamber

wards. This indicates that in these engines it is an inability to utilize the air which is actually available rather than the utilization of a large fraction of the air that is the cause of the relatively early appearance of colour in the exhaust.

10. Future Developments.

It is always dangerous to attempt to prophesy the trend of future developments, but it would seem that not very much further improvement in the basic efficiency of the compression-ignition engine is to be expected in future. If the results based upon the indicated performance which are given earlier in this chapter are studied in the light of the efficiency theoretically attainable after allowance has been made for the properties of the working fluid, the reader cannot fail to be struck by the very high relative efficiency which is being reached to-day. Thus the margin for future improvement is small, and, the true heat losses having been reduced to such small proportions, very little room is left for thermodynamical improvement, so that the author feels that very little can now be expected in that direction. In making this statement he does not mean that none of the engines being built to-day are incapable of thermodynamical improvement, as this is far from being the case, but that very little advance is to be expected beyond the best figures obtained to-day.

On the basis of the brake horse-power there is certainly more room for improvement, but to obtain this improvement we must look to the mechanical efficiency of the engine. The mechanical efficiency of the engine can be improved in two ways, by reducing the friction losses of the engine or by increasing its output, thus reducing the friction losses to a smaller proportion of the gross output. It is somewhat difficult to suggest means whereby the former can be achieved, but by careful design to ensure the maximum rigidity, the elimination of all unnecessary weight in the moving parts, and the employment of careful workmanship, something might be gained. Any reduction in friction losses will make itself felt to an increasing extent as the load is reduced, and as most engines are operated at fractional loads for by far the largest part of their working life, quite a small improvement will have a noticeable effect and is worth striving for. The second method appears to offer rather more scope, or perhaps it would be more correct to say that there is still some scope for producing an increase in the maximum output which can be obtained from a given cylinder capacity. Two methods are available for increasing the output: (1) by utilizing a greater proportion of the air already in the cylinder, and (2) by increasing the quantity of air in the cylinder.

Unfortunately, the utilizing of a greater proportion of the air results in a decrease in the efficiency of the cycle, and this decrease is usually more rapid than the increase in the mechanical efficiency produced by the improved output, so that although a greater power may be obtained from the engine, it is at some sacrifice of the specific consumption. This, however, is not necessarily a serious disadvantage, for although we may not have succeeded in improving the net thermal efficiency of

the engine, we have succeeded in obtaining a greater specific output. At the same time the fuel consumption under average conditions of running will remain unchanged and the average fuel consumption under service conditions will not be materially increased.

An increase in output obtained from an improvement in breathing capacity of the engine provides an opportunity for gaining an improvement in fuel consumption at all loads throughout the engine's useful range, because for the same degree of air utilization the efficiency of the cycle will remain unaltered and at the same time a higher mean pressure will be developed. This means that for any given mean pressure the degree of air utilization will be less and the efficiency of the cycle at any given output will be greater than was the case before the breathing of the engine was improved. At the same time the maximum output of the engine has been increased without any sacrifice of specific consumption. Accordingly, the greatest attention should be paid to the breathing capacity of the engine, especially to the avoidance of any unnecessary heating of the air during induction, because not only does this heating reduce the weight of air drawn into the engine, but it increases the temperature throughout the whole cycle and so decreases the efficiency. A good deal of heat is often picked up by the air before it actually reaches the engine intake, and quite a bit can be gained by avoiding any such heating. It is well to remember that approximately 1 per cent of the weight of air is lost for every 3° C. temperature rise of the air above that of the prevailing atmospheric temperature.

An obvious method of increasing the weight of air received by the engine is to utilize forced induction, i.e. by supercharging. This is a method which, under certain circumstances, offers considerable advantages, and, provided that the cost of the additional air is not too great, enables a large increase in output to be obtained with some improvement in fuel consumption. Supercharging is a subject which is beginning to attract a good deal of attention; properly applied, it has some very decided advantages and is certain to have extensive applications in the future. It is, however, a subject which cannot be dealt with adequately in the space available, and as it is really a future development the author prefers to omit further discussion of the subject.

The author is of the opinion that industry will continue to have a place for both the open and the indirect type of combustion chamber. The former will hold the field where fuel consumption is held to be of supreme importance. The latter, on account of its more catholic taste in fuels, will certainly continue to have a wide field of application, and with a further reduction in mechanical losses a point may eventually be reached where the difference in fuel consumption between the two types will be so small as to be of little account. Already in-

stances can be cited where consumptions of the order of 0·38 lb. per B.H.P. hr. are obtained, a figure which is at least as good as that obtained from many open combustion chamber engines. Such figures, coupled with a high specific output and the ability to utilize a wide range of fuels, some of them at a somewhat lower cost, are very strong points in favour of the indirect type of chamber. At the same time the specific output from the open chamber is certain to be increased by further development. This, as mentioned above, will mean some increase in specific consumption at the higher outputs, so that it would seem that ultimately the difference between the two types will be small and may eventually disappear, although the advantage of ability to utilize a wide range of fuels will be somewhat difficult to attain for the open chamber.

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